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POWER'S
PRACTICAL REFRIGERATION

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POWER'S
PRACTICAL REFRIGERATION

COMPILED BY THE
EDITORIAL STAFF OF POWER

FIRST EDITION

PUBLISHED BY
POWER

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PREFACE

The volume of refrigerating facts has been produced in answer to repeated requests by Power's subscribers for a book dealing with the practical side of refrigeration and still including the essential laws covering the production of refrigeration.

In the main the contents are made up of articles which have appeared in Power's columns and which were of particular value to engineers. Considerable space has been devoted to the subject of indicating the ammonia compressor since a more general use of the indicator would do much to improve the efficiency of operation.

The courtesy of Longmans Green and Co., and W. A. Goodnow, M. E. has enabled us to reproduce in this book the Ammonia Tables which appear in the final chapters.

It is *Power's* belief that every engineer will find in this volume something of benefit. If this is true then *Power's* ambition to be of service has been fulfilled.

THE EDITORS.

NEW YORK,
October, 1921.

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POWER'S PRACTICAL REFRIGERATION

SECTION I

REFRIGERATION MADE SIMPLE

CHAPTER I

THE REFRIGERATING PLANT AND THE STEAM PLANT

Mechanical refrigeration, whether it consists of the maintenance of a state of coldness in a storage room or of the production of ice, is, in all respects, a thermodynamic or heat process. The same cycle of alternating absorption and rejection of heat that occurs in the steam plant takes place in the refrigerating plant, and only differs in that the process is the reverse of the steam-engine cycle.

In Fig. 1 is shown a schematic layout of a condensing steam plant. The water in the boiler is supplied with heat from the coal burning on the grate. Only one heat unit to a pound of water is required to raise the temperature 1 deg. F. As a consequence, the rise in temperature is rapid until 212 deg. F., the boiling point at atmospheric pressure, is reached, when, if the boiler is open to the atmosphere, the rise in temperature appears to stop (the heat contained in the water is called the heat of the liquid, h), and no further rise is noticed until 970.4 heat units, known as the "latent heat of evaporation," have been added to each pound of boiling water. As this heat is absorbed the water turns into steam, and, if the steam is not allowed to escape, the pressure will rise. When pressure develops, a peculiar thing happens. The boiling point of the water is no longer

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at 212 deg., but at a definite temperature corresponding to the existing pressure. If the pressure were 150-lb. gage and heat were added, the water would boil when it reached a temperature of 366 deg. F., or, in other words, would evaporate from and at 366 deg. F., the water now contains more heat of the liquid h than when at 212 deg. F., and the latent heat would be reduced to 857 heat units. The values of h and L for any pressure may be found in the steam tables. Any temperature below 366 deg. would condense the steam at that pressure.

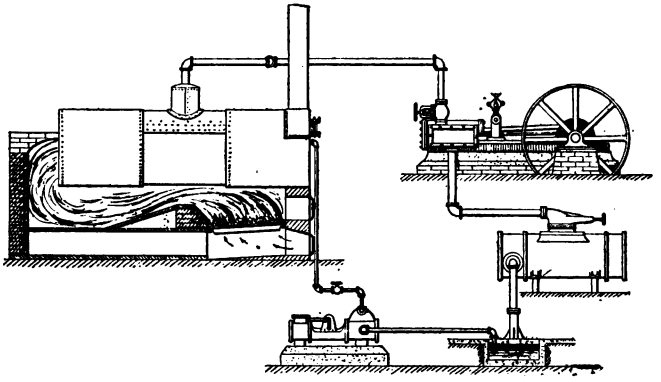


FIG. 1.—Schematic steam plant.

From the boiler the steam passes through the engine where it expands and gives up some of its energy in the form of work. Passing on to the condenser, cooling water is supplied to condense the steam into water which is returned to the boiler by means of the feed pump. This pump requires a certain amount of energy to drive it. To obtain the most economical results, the water should be returned to the boiler as hot as possible. To make the process ideal, the water should have the same temperature as in the hot well.

The cycle passed through by the water in being converted into steam, expanded in the engine cylinder, condensed and returned to the boiler may be represented by the diagram in Fig. 2. The water entering the boiler is at the condition represented by the point B. Heat is added and the water is heated and attains the pressure con-

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dition at *E*. Further addition of heat causes evaporation which steam flows into the engine cylinder and fills it to the point of cutoff at *F*. The steam then expands along *FH*. If the expansion were complete to the back pressure carried,

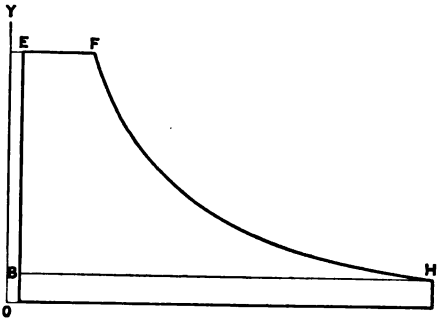


FIG. 2.—Steam cycle diagram.

the volume would be indicated by *H* in Fig. 2. This would entail too large a cylinder for the power developed. Consequently the exhaust valve opens at *H* Fig. 2*a*, and the steam is exhausted along *HS* and is condensed and reduced in volume along *SB*; whereupon it is again forced into the boiler. This is

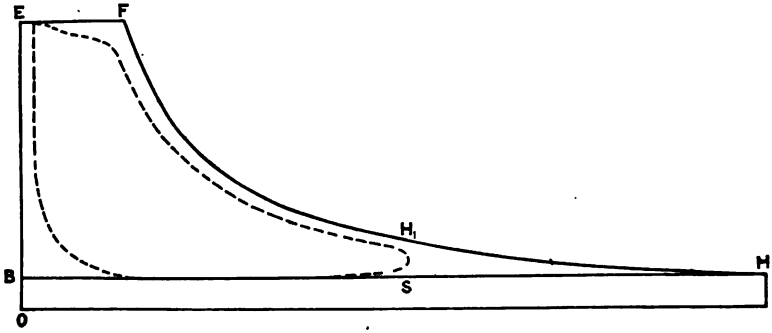


FIG. 2*a*.—Steam cycle diagram with incomplete expansion.

to all outward purposes identical with the ordinary steam-indicator diagram. Such a steam cycle with complete expansion is called the Rankine Cycle and the efficiency of an engine is usually stated in terms of its ratio to the efficiency

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of this ideal engine. Due to the cylinder condensation, wire drawing, compression, etc. an actual indicator diagram fails to exactly follow this diagram in Fig. 2*a*. Instead a diagram somewhat as shown by the dotted lines in Fig. 2*a* is secured.

REFRIGERATION CYCLES

It is apparent that water in being increased in temperature and evaporated absorbs heat from its surroundings, in fact from the boiler furnace. If a substance would boil at a very low temperature while at a pressure within range of practical machinery, it would absorb heat from its surroundings as long as the latter's temperature was slightly above the temperature of evaporation. Since the boiling point is always higher at increased pressure the substance under discussion after evaporation might be compressed to a higher pressure, and then if it is cooled below the boiling temperature at the compression pressure the substance would be condensed. It might then by the application of more and lower temperature cooling be reduced still lower in temperature. Now if this substance is allowed to flow through a valve into a container at a very low pressure it would instantly evaporate. In boiling, as before stated, heat is absorbed. Some of this heat of evaporation, L, would be drawn from the remaining liquid, reducing it in temperature to the temperature corresponding to the container pressure. More heat would be taken up by the boiling liquid from the surroundings. In this way a refrigeration effect would be secured. Many substances do have boiling temperatures which make them adaptable as refrigerating agents as will be outlined later. Ammonia, however, is the chief refrigerating medium in present-day use. Both compression and absorption systems are in general favor.

The ammonia-compression refrigerating plant is represented by the diagram in Fig. 3. Here the ammonia liquid is in the receiver at a temperature of around 70 deg. and 160 lb. pressure. In passing through the regulating valve its pressure is reduced to that of the evaporating coils, say 15-lb. gage. Since am-

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monia will evaporate, or boil, when at any temperature above zero F. when under a pressure of 15-lb. gage, or 30-lb. absolute, the ammonia liquid as it enters the coils immediately boils or springs into vapor. To do this, heat must be applied. Although the liquid ammonia contains the heat of the liquid corresponding to the receiver pressure which is used up in the evaporating process most of the heat must be absorbed from the brine or from whatever substance is around the pipe coil. In the absorption of this heat the temperature of the surrounding body necessarily is lowered.

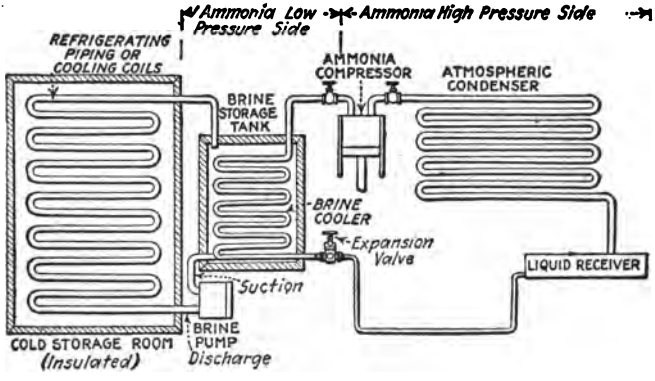


FIG. 3.—Schematic compression refrigerating cycle.

The ammonia gas is then drawn into the compressor and compressed to some 160-lb. or more per square inch. Since work is done on the gas in compressing, it increases in temperature. Some of this heat is absorbed by the compressor cooling jacket; the gas, however, is fairly hot when it leaves the compressor and in a gaseous and superheated condition. The cooling water in the condenser absorbs enough heat to lower the temperature of the gas to around 70 deg. Since this temperature is below the temperature at which ammonia boils while under a pressure such as exists in the discharge line, the ammonia gas liquefies and flows into the receiver, and from this point through the expansion valve into the coil again.

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The refrigeration cycle may well be represented by a diagram similar to the steam-cycle diagram in Fig. 2. The refrigeration cycle is shown in Fig. 5. Here we can start with the liquid ammonia in the liquid receiver at a pressure represented by the point *E*, with a corresponding temperature which can be obtained from the ammonia tables. As the liquid passes through the expansion valve it arrives at the pressure

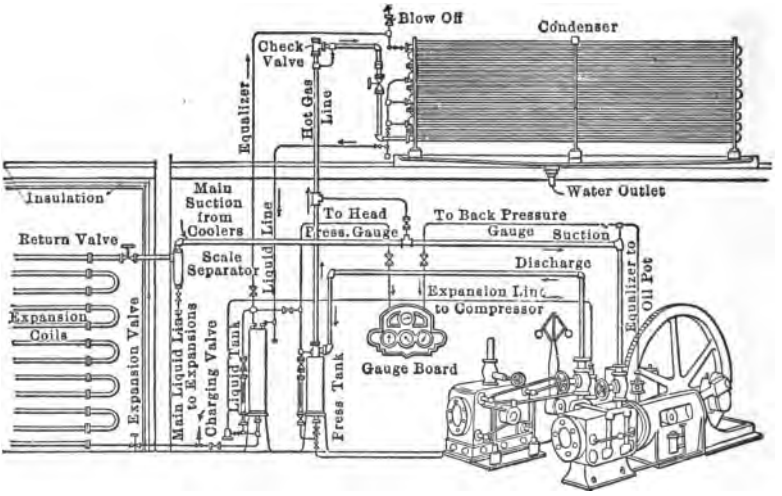


FIG. 4.—Typical compression refrigeration system.

point *B*. Since this pressure and temperature is below the boiling point of ammonia the liquid absorbing heat vaporizes and the volume increases along the line *BH* which is at the suction or coil pressure; entering the compressor it is compressed along the line *HF*, increasing in pressure and temperature while becoming more dense. At *F* it is discharged along *FE* into the condenser which removes the heat of vaporization and condenses it into liquid ammonia with a volume represented by the distance the point *E* is from the line *OY* and at a pressure *OE*. The cycle is now complete. It is evident that this is the reverse of the steam cycle. This is the cycle of all compression systems regardless of the media

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used. A typical compression system appears in Fig. 4. The ammonia-absorption system which is discussed in Chap. VIII while radically different in structural features operates on the same principle of alternate absorption and rejection of heat.

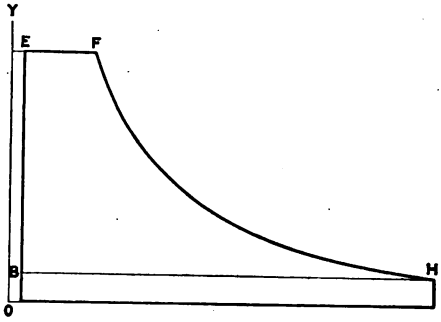


FIG. 5.—Refrigerating cycle diagram.

ENGINEERING TERMS

Before going very deeply into the events taking place in a refrigeration cycle, certain laws and enginering terms must be understood. The principal ones are as follows:

BOYLE'S LAW

Boyle's Law states that the volume of a given weight of gas when confined in a vessel varies inversely as the pressure if the temperature is kept constant. This can be put in the formula

$$PV = \text{a constant}$$

CHARLES' LAWS

Two principles of the action of a gas are termed Charles' Laws. The first one declares that under constant pressure the volume of a gas varies directly with the absolute temperature or

$$\frac{V_1}{V_2} = \frac{T_1}{T_2}$$

The second law states that with the volume constant the temperature varies with the absolute temperature or

$$\frac{P_1}{P_2} = \frac{T_1}{T_2}.$$

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ABSOLUTE PRESSURE

The pressure registered by a steam gage or similar device does not represent the total pressure acting on an object but represents the pressure above the pressure due to the weight of the atmosphere. The weight of a column of air reaching from the earth to the upper limits of the atmosphere and acting on a surface of 1 sq. in. has been found to be approximately 14.7 lb., varying with the altitude and with the weather conditions. The gage pressure then represents a pressure 14.7 lb. less than the total pressure. The latter pressure (gage pressure plus 14.7 lb.) is termed the absolute pressure and is used in most engineering calculations.

ABSOLUTE TEMPERATURE

If a gas thermometer has the freezing, 32 deg. F., and boiling, 212 deg. F., temperatures marked on it and these divided into 180 divisions, it is found that for every decrease of 1 deg. the length of the column in the thermometer decreases $\frac{1}{493}$ part of its total length from the bulb to the freezing point. It is conceivable, then, that, if the temperature was reduced 493 deg. below freezing point, the volume of the gas in the tube would be zero. This is the condition when the entire volume of a substance would disappear, and it is termed absolute zero, being 461 deg. below the Fahrenheit zero-point.

HEAT

Heat is assumed to be a condition of vibration of the molecules making up a substance, whether it be liquid, solid or gaseous. We know that when a chemical action takes place heat results; as an example, in burning coal in a boiler furnace, which is a chemical action, the velocities of the molecules of the water become so rapid as to cause them to separate and form steam. The abstract theory of heat is entirely too complicated for ordinary use. The engineer is actually interested in the result of heat rather than the causes.

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BRITISH THERMAL UNIT

A basis upon which to measure heat or rather its result is necessary. The amount of heat required to raise 1 lb. of water from 32 deg. F. to 212 deg. F. under atmospheric pressure has been measured. The heat unit used in engineering is $\frac{1}{180}$ of this amount of heat and is called the *British Thermal Unit.* It is then the average heat necessary to raise 1 lb. of water 1 deg. anywhere from 32 to 212 deg. F. It is usually abbreviated B.t.u.

HEAT OF THE LIQUID

If heat is added to a liquid which is under some stated pressure and is at an initial temperature T_o, the heat absorbed will cause the temperature of the liquid to rise until it attains a value dependent upon the pressure carried, whereupon any additional heat will cause the liquid to boil or vaporize. The heat required to raise the liquid to this boiling point is termed the *heat of the liquid,* usually denoted by h. Since some temperature base must be used, it is customary to take 32 deg. F. as T_o.

LATENT HEAT OF VAPORIZATION

After the boiling point of a liquid is reached any further addition of heat will cause part of the liquid to vaporize, the process continuing until the entire amount of liquid turns into vapor; the temperature and pressure remaining the same. This heat is called the *Latent Heat of Vaporization* (L). This heat is used up in the internal work of overcoming the cohesion of the liquid molecules and in doing the external work of increasing the volume of the combined liquid and vapor. It is evident that if the pressure remains constant the space occupied by the vapor and liquid must increase.

SPECIFIC HEAT

If a gas is confined in a receptacle and is heated the pressure will, of course, rise. If the volume is kept constant, the heat is said to be applied at constant volume. The amount of heat

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necessary to raise 1 lb. of gas 1 deg. under this condition of constant volume is termed the *Specific Heat at Constant Volume*, and is usually written C_v.

If the pressure is kept constant by allowing the volume to increase, as in a cylinder with a movable piston, the heat added per pound per degree Fahrenheit temperature increase is the *Specific Heat at Constant Pressure*, written C_p. It is apparent that, if the volume is allowed to increase, the gas does work on the movable piston. Work is done, and the heat added must not only increase the temperature 1 deg. but must also perform work. The specific heat at constant pressure, C_p, is then greater than the specific heat at constant volume, C_v.

ISOTHERMAL EXPANSION AND COMPRESSION

If the temperature of a gas is kept constant while being compressed heat must be removed. Likewise, if the gas is expanded, as behind a piston, heat must be added to keep the temperature at the same value. Such action is an isothermal one; *i.e.*, at constant temperature, and the relation $PV =$ Constant $= C$ holds.

ADIABATIC EXPANSION

If a gas expands without receiving any heat the work done by the gas during expansion is at the expense of the heat in the gas. As a consequence the temperature falls. Conversely, if during adiabatic compression no heat is removed the temperature rises. The pressure and volume with adiabatic expansion or compression have the relation

$$PV^N = C = \text{Constant},$$

Where the exponent N is equal to the ratio of the two specific heats C_p and C_v of the gas. The following relation also holds good

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

and

$$P_1V_1^N = P_2V_2^N$$

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THE HORSEPOWER

If a weight of 1 lb. is lifted 1 ft. an amount of work represented by W, the weight, and S, the height, is performed. This is equal to WS or one foot-pound. A horsepower is the performance of 33,000 ft.-lb. in 1 min. With an engine the weight is seldom lifted vertically but consists of the pull on the belt. The space S is represented by the belt travel per minute or $2\pi FRN$. The horsepower is then $2\pi FRN$, where $2\pi R$ is the circumference of the belt wheel, N the number of revolutions per minute, and F the belt pull (difference between the tensions of the tight and slack sides of the belt).

REFRIGERATION CYCLE

In the first part of this chapter an attempt was made to show what refrigeration really is and what means are resorted to to achieve this end. While any substance which boils at temperatures within the usual ranges may be employed in its practical application in appreciable capacity it is necessary to utilize a vapor like ammonia, which, in changing from a liquid to a gaseous state (at the proper pressure), would boil at a relatively low temperature. Also it is necessary to use the same refrigerant over and over again, which means that a pump or compressor would have to be provided to increase the pressure sufficiently high that the gas might be condensed at atmospheric temperature. This process was shown to be similar to the steam-engine cycle, only with the events in the reverse order. The object of the gas pump, or compressor, is simply and solely to increase the pressure, thereby making it possible to condense the compressed ammonia by cooling with water at 60, 70 or 80 deg. F. temperature.

THE HIGH-PRESSURE SIDE

The reader should understand the reason for the use of the compressor—what it accomplishes and what is expected of it. There is really no unusual feature about it. The dry-air vacuum pump draws in the rarefied air at the pressure maintained in the surface condenser and pumps it up to an amount

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a little in excess of atmospheric pressure, when it is exhausted out of the condenser. The deep-well pump takes water at the level in the well and lifts it up to the top or overflow. In each case work is done on the substance, it is lifted in level (pressure level in one case and datum level in the other). In the case of ammonia we are able to lift it in still another sense —in the temperature level, because by increasing the pressure we can condense the vapor at a relatively high temperature.

As in compressing air, the ammonia gas becomes hot during compression and the discharge gas has a pressure of 125 to 185 lb. per square inch and a temperature of between 150 and 250 deg. F. This compressed hot gas is allowed to leave the compressor and pass into a kind of aftercooler, which is a coil of pipe suitably water-cooled so that finally the temperature of the gas is reduced down to within 5 to 10 deg. of the initial temperature of the water. From what has already been said it is clear that long before this has occurred condensation of ammonia has been completed and we have a liquid ammonia under a pressure approximately that of the discharge from the compressor. This liquid is collected into a drum, or header (called a liquid receiver), and is now ready for a second round through the refrigerating coils. The compressor, condenser and receiver are often called the high side.

The liquid ammonia stored in the receiver is under pressure and will flow through the pipe line to the cold-storage room similar in action to the manner of water in a pipe. At the cold-storage room, or brine tank, there is a valve called an expansion valve. It is not an expansion valve at all; it is only a pressure-reducing valve. It has been shown that to get a boiling temperature of zero deg. F., the ammonia must be kept at 15-lb. gage. So then, the expansion valve allows the pressure to drop from 150 lb. to 15 lb. and the liquid squirts into the coils like water under pressure through a sand hole in a pipe fitting. If the piping is properly arranged and proportioned, it will immediately begin to boil because of absorbing heat through the pipe coils from the air in the room, in consequence of which it becomes vaporized. As a vapor the ammonia has little refrigerating value, and it is allowed to pass

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out of the coils promptly and return to the compressor, which compresses it again.

THE REFRIGERATION CYCLE

And so it will be seen that the refrigeration cycle has three parts—the refrigerating coils, the compressor and the condenser. The refrigerating coils in action are much like the boiler in a steam-engine plant. The coils of pipe are so arranged as to be in contact with relatively hot air or brine. This heats the pipes and boils the ammonia, vigorously or not, depending upon the temperature difference between the air temperature in the cold-storage room and the boiling temperature of the ammonia. Now, in a steam boiler, if boiling took place the steam would accumulate, and unless the engine or some other apparatus took the steam away as fast as it was generated, the pressure would rise until finally the safety valve would blow. On the other hand, if the steam were used too fast the steam pressure would drop and the temperature would be lowered in consequence. That the pressure may stay constant in a steam boiler we have to have means of generating just the same amount of steam as is used each minute.

Likewise, the refrigerating plant must be so ordered that the vapor boiled from the refrigerating coils will be just enough for the compressor. If the boiling action is too rapid and more vapor is formed than can be pumped by the compressor, the boiling *pressure* will increase and the boiling *temperature* will likewise increase, perhaps to a point where satisfactory results will not be obtained. On the other hand, too high a speed of the compressor will lower the boiling pressure and temperature and poor results again will be encountered. The cubic feet of piston displacement of the compressor and the evaporating system must be inter-related.

For example, suppose we have a room to be held at 36 deg. F. and 500 sq. ft. of cooling coils is supplied. The design allows a suction pressure of 37½ lb., which corresponds to 24 deg. F. boiling temperature of the ammonia. The temperature difference on the two sides of the pipe is 12 deg. F., and in consequence a certain amount of heat will flow through the

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pipe from the air to the ammonia. The amount will vary directly in proportion to the product of the area of the pipe and the temperature difference. The result is that ammonia is evaporated and it will be given off at the rate of 1.8 cu. ft. per minute.

It may not be clear to everyone that a gas or vapor has weight. Yet in a steam engine the steam condensed may be weighed, and we know that it requires 15, 20 or 30 lb. of steam to produce a horsepower for 1 hr. Likewise, air has weight, as we know that the atmosphere presses down with a pressure of 14 lb. per sq. in. So each pound of ammonia evaporated will occupy a certain particular volume, depending on the pressure. For instance, 1 lb. of ammonia at

200-lb. gage will occupy 1.40 cu. ft.
150-lb. gage will occupy 1.81 cu. ft.
100-lb. gage will occupy 2.57 cu. ft.
30-lb. gage will occupy 6.25 cu. ft.
20-lb. gage will occupy 7.93 cu. ft.
10-lb. gage will occupy 11.0 cu. ft.
0-lb. gage will occupy 17.6 cu. ft.

REFRIGERATION A MATTER OF HEAT TRANSFER

In the steam boiler it is usual to specify 10 sq. ft. of heating surface per boiler horsepower, and a boiler horsepower as the evaporation of 34½ lb. of water from 212 deg. to dry steam at 212 deg. F. In refrigeration we cannot proceed just like that. The refrigerating coils are really a boiler, but conditions are not always the same. For instance, we may design the coils to have 10 deg., 15 deg., or 20 deg. or more difference in temperature between the inside and the outside of the pipes. The greater the difference in temperature the more capacity may be obtained from the pipe refrigerating surface. Again, the pipe surface may be wet or frosted slightly, or heavily coated with ice. It is not reasonable to expect that the same amount of boiling action will occur in each of these cases, nor in the case where the inside of the pipe is filled with oil. In other words, there is a so-called coefficient of heat transmission for refrigerating piping as there is in all heat transmission piping.

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As refrigeration is almost entirely a matter of heat transfer, it is desirable to be able to calculate how this takes place. The equation which we use is

B.t.u. = area of surface in square feet × coefficient of heat transfer × the temperature difference in degrees F. on the two sides of the surface.

The coefficient of heat transfer has to be found experimentally or by experience on similar kinds of work. The following problem will show the method of calculation. Suppose we have a direct-expansion pipe composed of 1,200 lin. ft. of 1¼-in. pipe. The ammonia boils at 25-lb. gage (12.2 deg. F.), and the room temperature is to be maintained at 34 deg. F. It is required to find the refrigeration to be supplied. The usual value of the coefficient of heat transfer under these conditions is 2.0 B.t.u. per hour, then

$$\text{Total B.t.u.} = \frac{1{,}200}{2.3} \times 2 \times (34 - 12.2) = 22{,}748 \text{ per hour.}$$

Steam boiler capacity is defined as the rate of *heating*, or at 33,479 B.t.u. per hour. In like manner we define the standard of refrigeration (the ton) as the rate of *cooling*, but in this case at the rate of 12,000 B.t.u. per hour (or 200 B.t.u. per minute). So, in the foregoing problem the tonnage will be $\frac{23{,}200}{12{,}000} = 1.89$ tons of refrigeration, and a machine would have to be selected which would compress enough vapor to supply this amount of refrigeration. How this may be determined for various conditions will be shown later.

However, in the final analysis the real points to be considered are the horsepower input per ton of refrigeration and the cubic feet of piston displacement of the compressor per ton. All other factors fade in comparison unless the conditions are ultra-extreme. Of course we desire to produce refrigeration at as small cost of power as we can, and this cost of power is mostly in the demands of the compressor. The compressor pumps the gas from the refrigerating coils to the condenser, and the condenser pressure is determined by the temperature and the amount of cooling water used. Again, as a rule, the *volume* of gas handled affects not only

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the size and cost of the compressor, but of the high- and low-pressure sides as well.

POWER USED PER TON OF REFRIGERATION

It so happens that, contrary to the expectation of most engineers, the power required by the different refrigerating media per unit of refrigeration is about equal. One could expect that carbon dioxide, working as it does with a pressure range of about 550 lb., would require more power input than

TABLE 1.—SOME PROPERTIES OF REFRIGERANTS

	Latent heat at zero deg. F., B.t.u.	Net refrigeration effect at zero deg. F.	Volume gas pumped per 100,000 B.t.u. refrigeration, cubic feet	Column 3 with carbon dioxide as unity	Approximate condenser pressure, pound	Approximate suction pressure, pound
Sulphur dioxide...	171.8	148.7	4,970	12.4	35.0	− 5.0
Ammonia.....	572.1	496.4	1,848	4.61	114.5	15.25
Carbon dioxide.....	117.0	73.1	401	1.0	834.0	291.0
Ethyl chloride....	149.4	119.4	15,720	39.3	5.5	−10.6

All pressures are gage.

would, say, sulphur dioxide, which works between the limits of −5 lb. and 35 lb. or only 40-lb. total pressure. However, the volumes to be pumped are to be considered, and there are 12.4 times as much sulphur dioxide as carbon dioxide to be pumped for equal refrigerating effect. The net result is that carbon dioxide requires a little more power than does ammonia or sulphur dioxide. In fact, ammonia leads slightly in this respect, but the net result is so close that the matter of the horsepower per ton of refrigeration for usual (standard) operating conditions need not be considered when deciding on one or the other refrigerant.

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CHAPTER II

THE AMMONIA COMPRESSOR

In the preceding chapter it was shown that a refrigerating cycle was much like a steam-engine cycle, only that it was a cycle with the events in the opposite order. Also, it was shown that the refrigerating coils acted similarly to a steam boiler, but that the capacity of the coils depended on local conditions of operation. The compressor is the chief part of the machine requiring an input of power. Whereas inefficient condensers or refrigerating piping might prevent obtaining capacity, yet the compressor is the place where economy of operation may be looked for.

COMPARISON WITH THE AIR COMPRESSOR

The problem of compressing ammonia is so like that of compressing air that the same machine has been used for either substance.

One difference is to be found in that single-stage compression is used with ammonia except in isolated cases—although one may venture to say that two-stage compression will be used more in the future. The use of the single stage is surprising when one remembers that in air compression multi-stage generally is employed if the discharge pressure is 70 lb. or more. To understand the problem of compressing ammonia a study of the valve action will be made, as shown in the indicator diagram, Fig. 6.

The reader will remember that the indicator diagram is simply an indication of the pressure in the cylinder for each and every portion of the piston. By means of special apparatus we are able to record to a special (known) scale the pressure inside the cylinder during the entire cycle of suction, compression and discharge of the gas. If the entrance into the cylin-

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der is constricted, then the gas will have difficulty in getting in and the result is a partial vacuum formed during the suction stroke. If the discharge passages are constricted, there will be difficulty in getting the gas out of the cylinder promptly, and in forcing it out the piston has to increase the discharge pressure. Let us see how these points are represented on the

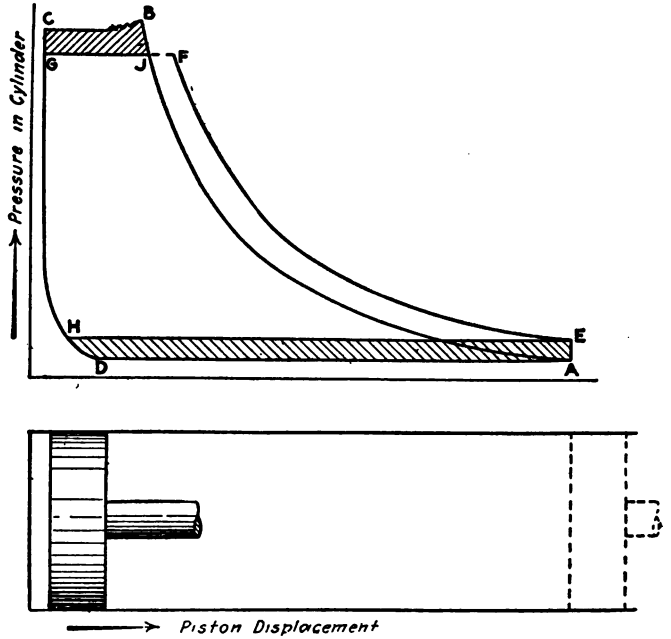


FIG. 6.—Compressor indicator diagram.

diagram, remembering that every square inch of area on the indicator diagram represents a certain number of foot-pounds of work done on the gas by the piston, and so by the engine or electric motor driving the piston.

EFFECTS OF VALVES AND PORTS

There are two separate indicator diagrams shown in Fig. 6. First we will discuss the one shown by the letters *ABCD*. This diagram represents the effect of suction and discharge valves

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and passages that are too small for the size and speed of the piston. The ammonia gas begins to enter the cylinder at the point D and continues to the end of the stroke represented by the point A. On the return stroke the suction valve closes and the entrapped gas is compressed along the line AB until the discharge pressure B is reached when the discharge valve opens and the compressed gas leaves the cylinder into the discharge pipe and the condenser. As the piston begins to move toward the right, the gas in the clearance volume expands until a pressure equal to the ammonia pressure in the suction bends is reached or some lower pressure, depending on conditions that prevail. This point is shown by D when the suction valve again opens for passage of a new volume of suction ammonia.

Now, if the suction gas could have entered freely into the cylinder, the pressure would have been higher and the suction line would have been HE. Also, if the discharge out of the cylinder could have been free, the exit would have been shown by FG. In each case the cross-hatched area $DAEH$ and $JBCG$ is unnecessary work, but has to be supplied to get the gas into and out of the cylinder. In addition (as the suction pressure was less) the weight of ammonia pumped per revolution is much less, as shown partly by the length of the lines GF, GJ. It will be seen, if a study is made of ammonia-compressor design, that the main differences in each design (other than in safety of operation) are in an attempt to provide as free and unrestricted passage into and out of the cylinder as is possible for the type of construction that is used.

WEIGHT OF AMMONIA HANDLED BY COMPRESSOR

The weight of dry saturated gas (cubic feet per minute) that must be handled by the compression to produce 1 ton of refrigeration per 24 hr. is

$$W = \frac{200}{L - (h_1 - h_2)}$$

where

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W = desired weight of ammonia,
L = latent heat of evaporation at the evaporating temperature and pressure in the coils,
h_1 = heat of the liquid at the condenser pressure,
h_2 = heat of the liquid at the evaporating pressure.

Example? Condenser or head pressure 198-lb. absolute; suction (evaporating) pressure 30-lb. absolute.
Then from tables of the "Properties of Ammonia" on page 265.

L = 572.1 B.t.u. at 30 lb. absolute.
h_1 = 71.6 B.t.u.
h_2 = − 33.6 B.t.u.

and $$W = \frac{200}{(572.1) - (71.6 + 33.6)} = 0.43 - \text{lb.}$$

Since in figuring the compressor displacement per ton of refrigeration the volume rather than pounds of ammonia is easier to use, the formula for weight can be changed to volume in cubic feet per minute by multiplying W, the weight obtained by the specific volume of 1 lb. of ammonia at the suction pressure assuming dry suction—this can be obtained from the ammonia tables. In the example given the specific volume at 30-lb. absolute is 9.17 cu.ft. The piston displacement then becomes

$$9.17 \times 0.43 = 3.943 \text{ cu. ft. per minute per ton}$$

Volumetric efficiencies of compressors vary from 60 to 85 per cent, and in obtaining actual cylinder volumes this discrepancy must be taken into account. Builders guarantee 80 to 85 per cent volumetric efficiencies in machines of 50 tons capacity and over.

In compressing the ammonia vapor from the suction pressure up to the head or condenser pressure the work preformed by the piston raises the temperature of the ammonia vapor between the values corresponding to the suction and head pressures, providing the condition of the vapor is dry saturated. All this work goes into raising the heat content of the

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vapor. The horsepower to compress 1 lb. of ammonia vapor per minute is obtained by the formula

$$\text{Hp.} = \frac{H_s - H_2 \times 778}{33{,}000}$$

when

H_s = total heat of the superheated ammonia vapor at the heat pressure,

H_2 = total heat of the ammonia vapor at the suction pressure.

778 = the mechanical equivalent of 1 B.t.u. in foot-pounds.
33,000 = 1 hp. in foot-pounds per minute.

If, as is usual, the discharge vapor is in a superheated condition the total heat at the discharge temperature is not that shown in the tables for saturated ammonia but includes the heat of superheat which is equal to $(t_s - t_1)S$, where t_3 is the temperature of the ammonia vapor, as found by a thermometer t_1 is the temperature of saturated ammonia at the head pressure (this is given in the ammonia tables), S is the specific heat of superheated ammonia vapor which is generally taken as 0.55. The formula then becomes

$$\text{Hp.} = \frac{H_1 - H_2 + (t_s - t_1)\, s \times 778}{33{,}000}.$$

It is more convenient to find H_2 direct from the tables for saturated ammonia. After finding the number of pounds of ammonia required per minute to deliver 1 ton of refrigeration per 24 hr. the formula will give the horsepower per ton of refrigeration under any set condition of suction and head pressures.

The actual horsepower per ton of refrigeration varies from 1.25 to 2 hp. per ton—depending upon the size of compressor, pressures carried, etc. On an average a 6-ton machine requires 2 hp. per ton, a 25-ton machine, 1.54 hp. per ton, and a 50-ton machine, 1.27 hp. per ton. A 1,000-ton compressor at the Quincy Market, Boston, operating at 750 tons capacity, produced 1 ton of refrigeration on 0.9 hp. with a compound-condensing engine.

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MEAN PRESSURE IN AMMONIA COMPRESSOR

Table 1 below gives closely the mean pressure in the compressor cylinder. To be used when estimating pressure in absence of indicator diagrams.

TABLE II

Condenser pressure, pound-gage		103	115	127	139	153	168	184	200
Condenser temperature, degree Fahrenheit		65	70	75	80	85	90	100	105
Refrigerator (back) pressure, gage	Refrigerator temperature degree Fahrenheit								
4	−20	41.46	43.91	46.34	48.77	51.23	53.68	56.11	58.54
6	−15	42.72	45.38	47.90	50.74	53.40	56.08	58.86	61.40
9	−10	44.40	47.38	50.33	53.29	56.25	59.20	62.16	65.14
13	−5	45.86	49.15	52.42	55.70	58.97	62.25	65.53	68.81
16	0	46.94	50.56	54.16	57.78	61.40	65.00	68.62	72.22
20	5	47.74	51.73	55.70	59.68	63.67	67.66	71.62	75.61
24	10	48.04	52.40	56.77	61.13	65.51	69.86	74.24	78.59
28	15	47.88	52.67	57.44	62.23	67.02	71.81	76.60	81.39
33	20	47.08	52.30	57.53	62.75	67.98	73.23	78.46	83.68
39	25	45.06	51.34	57.05	62.50	68.46	74.17	79.88	85.58
45	30	43.16	49.71	55.92	62.14	68.35	74.56	80.77	86.98
51	35	40.52	47.26	54.02	60.76	68.52	74.28	81.02	87.78

From De La Vergne Catalog.

HORSEPOWER PER TON

The table below gives conservative figures on the horsepower required per ton of refrigeration with various suction

TABLE III.—HORSEPOWER PER TON OF REFRIGERATION

Condenser pressure pound gage	Suction pressure pound gage					
	5	10	15	20	25	30
145	2.02	1.69	1.40	1.26	1.11	0.99
155	2.11	1.76	1.48	1.38	1.19	1.05
165	2.21	1.85	1.57	1.41	1.25	1.13
175	2.34	1.94	1.65	1.48	1.34	1.18
185	2.48	2.03	1.75	1.57	1.40	1.27
195	2.55	2.14	1.84	1.67	1.48	1.33
205	2.69	2.24	1.91	1.75	1.55	1.40

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and discharge pressures. Volumetric efficiencies are also shown.

TABLE IV.—VOLUMETRIC EFFICIENCY OF AMMONIA COMPRESSORS

Condenser pressure, pound gage		Suction pressure pound per square inch gage				
		0	10	20	30	40
120	A.	0.77	0.83	0.87	0.89	0.91
	B.	0.60	0.70	0.77	0.81	0.84
	C.	0.52	0.65	0.72	0.77	0.80
160	A.	0.74	0.80	0.83	0.86	0.88
	B.	0.54	0.65	0.72	0.76	0.80
	C.	0.44	0.58	0.66	0.72	0.75
200	A.	0.71	0.77	0.81	0.84	0.86
	B.	0.49	0.61	0.68	0.72	0.75
	C.	0.37	0.52	0.62	0.67	0.71

A, No clearance. B, 4 per cent clearance. C, 6 per cent clearance.

QUANTITY OF AMMONIA FOR REFRIGERATION

It is a common mistake for operating men to take for granted that the cylinder of an ammonia compressor fills with vapor having the same temperature as that in the evaporating coils and suction line. To remove heat from brine in a tank or from produce in a room ammonia has to be evaporated. This evaporation is caused by the absorption of the heat by the ammonia. The greater the number of heat units carried by the ammonia the greater the quantity of refrigeration produced. Also the greater the weight of liquid ammonia changed into vapor the greater the refrigerating effect. The capacity of the plant depends on the amount of ammonia it is possible to take from the evaporator and put into the condenser. Therefore the compressor controls the capacity.

The weight of ammonia is the chief consideration. To remove a given amount of heat, a certain number of pounds

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of liquid ammonia must be evaporated. At 30-lb. absolute, 1 lb. of ammonia will require 572.1 B.t.u. to change it from liquid to vapor. This means that at that pressure 1 pound of ammonia is capable of absorbing and carrying 572.1 B.t.u.

If the liquid admitted to the evaporating coil has the same temperature as the vapor in the coil, the whole load of heat is taken from the substance to be refrigerated. In the case under discussion the temperature would be 0.1 deg. to correspond with 30-lb. absolute pressure, if the heat transfer were perfect.

In ordinary practice the temperature of the ammonia passing the expansion valve is not the same as the temperature in the evaporating coil, but is that of the cooling water flowing over the condenser. If the cooling water is 80 deg. and the ammonia is so handled that its temperature is 80 deg. at the expansion valve, the difference in the heat of the liquid at the two temperatures of 80 deg. and 0.1 deg. F. must be absorbed by the ammonia in evaporating before any heat is taken from the pipe walls. Then $L - (h_1 - h_2)$ is the heat absorbed from the cooling coil and is the refrigerating effect obtained in B.t.u. In this case $572.2 - (53.6 - [-33.6]) =$ 484.9 B.t.u. refrigeration per pound of ammonia, or for each 9.17 cu. ft. of ammonia vapor at 30-lb. suction pressure handled by the compressor.

If it were possible to have a perfect compressor and perfect insulation and means of removing the heat of compression, so that no heat was added to the vapor, 1 pound of ammonia would be sent to the condenser every time the piston displaced 9.17 cu.ft., but such perfect conditions are never met.

It is possible to have the suction line between the evaporator and the compressor so well insulated and the ammonia so handled that a thermometer inserted in the suction line close to the compressor will show a temperature of 0.1 deg. when the back pressure is 30-lb. absolute. However well-insulated and water-jacketed the compressor may be, the temperature of the ammonia will increase in its passage into the cylinder. The suction valve, piston, cylinder head and cylinder

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walls have a temperature much higher than the incoming ammonia.

The ammonia at the end of the suction stroke will not be a saturated vapor, but a superheated gas. Just what the degree of superheat will be will depend on the condition of the compressor. In a single-acting dry machine this temperature is generally 100 deg., at which each pound of superheated gas occupies 11.55 cu. ft. If the superheat should be 150 deg.—and it often is—the volume will be 12.65 cu. ft. to the pound.

GAS IS SUPERHEATED IN CYLINDER

As it is impossible to prevent superheating in the cylinder, the only thing to do is to keep the superheat as low as possible. Have the ammonia reach the compressor in the greatest degree of saturation permissible with the type of machine and the operating conditions. A thermometer inserted in the suction line close to the compressor should show a temperature corresponding to the suction pressure. If the machine is vertical, single-acting and has a full-length water jacket, the bottom two-thirds of the jacket should be filled with a mixture of granulated cork and cement if necessary. This will set hard, and water will not penetrate it; it is an excellent insulating material. Through the top third of the jacket as much water as possible should be circulated to remove the heat of compression. The water flowing through the jacket should at no time impart heat to the ammonia.

The effective displacement of a compressor is seldom over 85 per cent of the cubical content of the cylinder. The other 15 per cent is accounted for by reexpansion of the gas in the clearance spaces, and valve and piston leakage. At that rate, to pump 1 lb. of vapor at 30-lb. pressure and 0.1-deg. temperature, the cylinder area must be 9.17 cu. ft. plus 15 per cent, or 10.5455 cu. ft. But as the ammonia is superheated 100 deg., the area through which the piston must travel to send 1 lb. of ammonia to the condenser is 11.55 + (15 per cent of 11.55) = 13.2825 cu. ft. We have learned that it is practically impossible to prevent superheating.

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If each pound of ammonia is capable of taking 481.014 B.t.u. from the substance to be refrigerated, then $\frac{288,000}{481,014} = 598.75$ lb. of ammonia is pumped in 24 hr. to produce 1 ton of refrigeration under the conditions given. The weight handled per minute will be $\frac{598.7}{1,440} = 0.415$ lb.

To sum up, $13.2825 \times 0.415 = 5.492$ cu. ft. of piston displacement will be required per minute to produce 1 ton of refrigeration in 24 hr. when the temperature of the liquid ammonia is 80 deg. F. and the temperature in the evaporator is 0.1 deg.

CALCULATING THE AMMONIA REQUIRED IN A SYSTEM

To make the situation interesting assume that, the next time you are examined for a license, the examiner should pop this question: "Explain what the quantity of ammonia should be in the refrigeration system, and show how to calculate how much of the refrigerant should be circulated per hour for, say, a 50-ton plant carrying a back pressure of 25 lb. and a condenser pressure of 175 lb."

There must be enough ammonia in the system to insure a solid stream of liquid going through the expansion valve, and there should be enough vapor evaporated from the liquid to make the evaporating coils take up heat equal to the capacity of the plant. In addition there must be enough surplus to allow for loss by leakage and for possible pocketing of the liquid somewhere in the system. It is evident that one of the items needed is the volume of the evaporating coils.

This is easily found either by calculating or by the use of a table of pipe dimensions found in the appendix of this volume. Table V gives these dimensions for pipe of evaporating-coil sizes:

Suppose the 50-ton plant has 12,000 ft. of 2-in. pipe in the expansion coils. We have $120 \times 2.360 = 283.2$ cu. ft. of coil volume to be filled with gas at 25-lb. pressure. Now we

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need a table of properties of saturated ammonia to find the weight of a cubic foot of gas at 25 lb.

TABLE V.—PIPE DIMENSIONS FOR EVAPORATING COILS

Internal diameter, inch	Length per cubic foot	Cubic foot per 100-ft. length
¾	270.25	0.370
1	166.9	0.599
1¼	96.25	1.038
1½	70.66	1.415
2	42.91	2.360

Referring to Table 23 page 260, we find that at 25-lb. gage, 40-lb absolute pressure, ammonia weighs 0.1430 lb. per cubic foot therefore the weight of gas in the 12,000 ft. of 2-in. pipe will be 283.2 × 0.1430 = 40.49 lb. This quantity will do for the expansion coils, but ammonia is needed for the condenser, receiver, oil trap, etc., as well. Practice has shown that the following quantities, in addition to the amount required for the expansion coils give sufficient amounts for the systems.

AMMONIA REQUIRED IN EXCESS OF THAT NEEDED FOR THE EXPANSION COILS

Capacity of machine, tons	Pounds of ammonia	Capacity of machine tons	Pounds of ammonia
10	200	100	580
15	250	150	680
25	350	200	780
35	400	300	1,080
50	450	400	1,380
65	500		

As 40.49 lb. is needed for the expansion coils and 450 lb. for the compression side of the system, 490. 49 lb.—say five 100-lb. drums—is required for the working charge. Of course, during operation or when ammonia is added to the system, the receiver glass will show whether the charge is too little or

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too much. Also, with too little in the system, and on a heavy load, there will not be enough liquid to prevent gas from going through the expansion valve. When there is much too great a quantity of ammonia in the system, the receiver will show full and the liquid will begin to fill the condenser. If this condition is extreme the head pressure will rise.

INDICATING THE AMMONIA COMPRESSOR

The best-known guides in determining whether the mechanical condition of a compressor or an engine is what it should be are pressures, and the indicator offers the best means of plotting a graphical record of the relative magnitude of the successive pressures and volumes inside the cylinder.

It is to be regretted that the indicator is not more commonly used to its fullest advantage; often it is applied carelessly. The best indicator kept in the best condition and operated by the most careful and intelligent man produces diagrams that are none too good. Indicator practice requires great care, both of the instrument and its method of attachment, since so much depends on a small area.

Defective valves and leaky pistons in a compressor, if permitted to run in this condition for long, needlessly waste much money, and these conditions in the compressor are much more common than is generally supposed.

Figures 7 and 8 are reproductions of ammonia-compressor diagrams, and upon looking at either, as it is, what does it tell you?

It tells nothing. It is worthless, and that is the point to remember—that is, merely taking a diagram, if that is as far as the process goes, is a waste of time. To make diagrams of value, the engineer needs guides—something to compare them with that he knows is all right—and these are the adiabatic curve and the isothermal curve.

In adiabatic compression the gas neither gains nor loses heat during the process. The adiabatic curve represents this action. If the heat of compression is removed as fast as it is produced, the compression is said to be isothermal and the temperature remains constant.

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Anyone can recognize a curve in a line that is supposed to be straight, without a straight line for comparison but a variation in a compression or an expansion curve from the corresponding adiabatic or isothermal curve (which is only one of an indefinite number no two of which are alike because of their changing eccentricity) cannot be even approximately judged without the presence of the proper line for composition. Every engineer will admit the importance of this point, and one is safe in saying that less than ½ of 1 per cent of the compressor-indicator diagrams taken even have their compression lines compared with the appropriate theoretical curve.

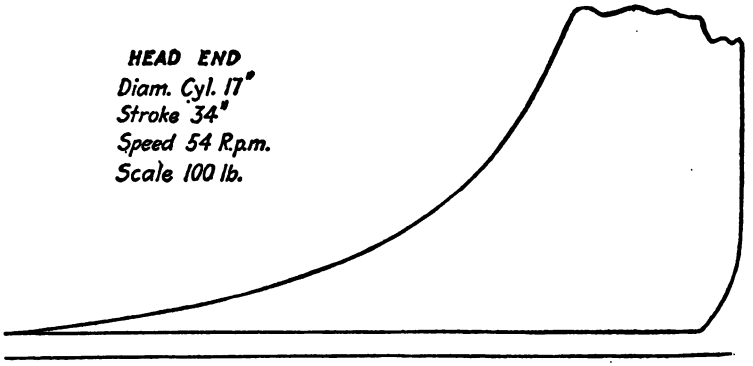


FIG. 7.—Ammonia indicator diagram.

In the average steam plant with an engine working under nearly constant conditions, the variation of the expansion line from the hyperbolic is usually less and more nearly constant than that of a compression line from the adiabatic, and this will be found to be so of almost any compressor, no matter how uniform the load. This is because the conditions in the steam cylinder vary comparatively slightly as to temperature and moisture, while those of ammonia in the compressor cylinder vary both in initial and final pressure every time a change is made in the expansion valves, speed of the machine varied, change in water temperature at the condensers, as well as the temperature and humidity of the atmosphere; performance is also subject to variations in humidity of the ammonia gas

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from the expansion coils. The fact that variations of humidity as well as pressures have their effect upon the ammonia indicator diagram, together with the fact that the poppet type of valves, which are more or less liable to stick, close sluggishly or leak, is used in compressors, makes it far more important that for intelligent interpretation the theoretical curve be plotted on the card than that the hyperbolic curve be plotted on the steam-engine-indicator card.

These variations often make the diagrams not only worthless but also harmfully misleading to one who depends upon only a casual inspection of them, in trying to determine whether the compressor is giving the best results. But if these diagrams taken under varying conditions are properly studied and compared, much valuable information can be derived from them, and the conditions that are conducive to the best results determined.

Despite the value of the indicator, many compressors have no connections, etc. for attaching one, and a greater number of plants have no indicator; then, again, some plants have a complete equipment for indicating and yet the engineer never uses it.

The indicator diagram will tell nearly everything that could affect the successful operation of the compressor when the adiabatic and isothermal curves are plotted on it and compared with the curves made by the indicator. It takes time, study, and experimenting to be successful at this, but one will be amply repaid for the time so spent.

On the diagram, Fig. 8, the admission line, *D E*, is parallel to the atmospheric line. The compression line is a curve gently rising to the discharge line. The discharge line is generally peaked and wavy, due to discharge-valve vibration. The admission, or section-pressure, line and the discharge line should be joined by a nearly vertical line. The diagram should have a nearly square heel at *D*.

The adiabatic and isothermal curves are also shown and will be discussed later.

Referring to Fig. 7, a casual inspection does not reveal any serious defects. The clearance is small, and apparently

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everything looks all right. Taking up Fig. 8 in the same manner, this shows a little more reexpansion or clearance loss then should exist, otherwise nothing serious is noticed. Further than this no engineer can go without the aid of the adiabatic curve. It is impossible to tell by a mere inspection whether the compression curves are good or bad. Should the

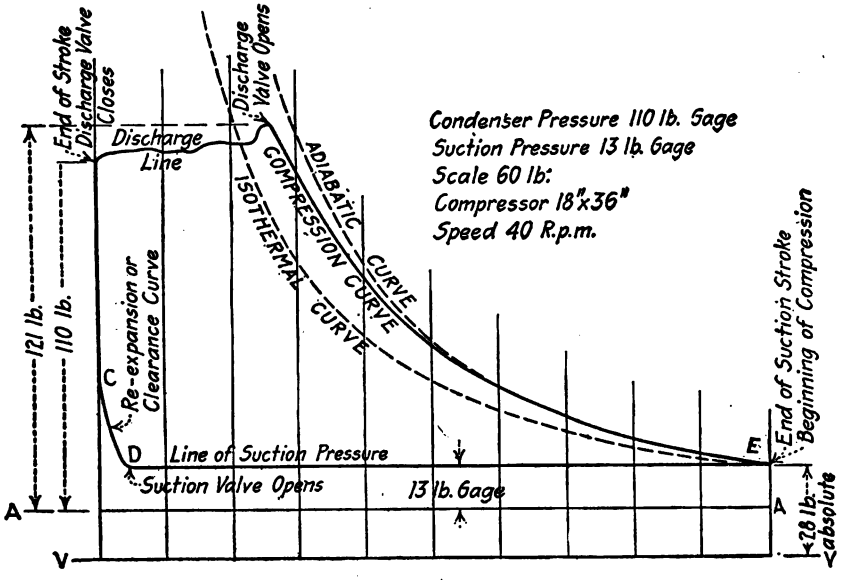


FIG. 8.—Typical ammonia diagram.

compression curve be very bad, it might be readily noticed; even then probably no one could pass an opinion upon it that would be of real value.

The isothermal line is also a guide, though not as good as the adiabatic line, but should both lines be plotted. It takes but a few minutes longer and is more than worth the time and trouble, as it makes the comparison more complete and interesting.

Figures 7 are again reproduced in Fig. 9 having the adiabatic and isothermal lines plotted. This diagram, which before apparently showed no serious defects

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is here shown to be very bad. This machine was needlessly wasting much money, though its poor condition was not suspected, because the engineer lacked the knowledge for applying the proper curves to the diagram to show him the true condition of the compressor.

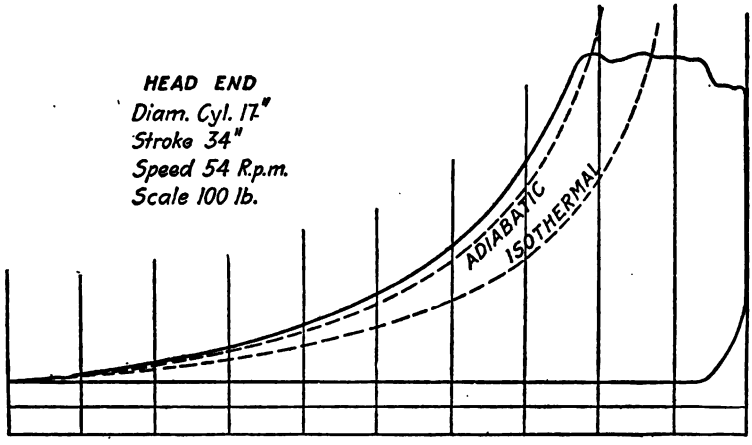


FIG. 9.—Fig. 8 with adiabatic and isothermal lines added.

An adiabatic line is a curve that represents the adiabatic expansion or compression of a gas or vapor. Adiabatic expansion or compression is the expansion or compression of a gas or vapor without loss or gain of heat. During compression the pressure gradually increases until the discharge valve opens; the curve to which this change of pressure corresponds most closely is the adiabatic curve. The law for the adiabatic compression of ammonia gas is similar to that for air, but the coefficient n in the familiar formula $pv^n = p_1v_1^n$ has a value of 1.3 for ammonia, instead of 1.405 ,as for air, and the formula is therefore $pv^{1.3} = P.V.^{1.3}$ where p equals initial pressure, v equals initial volume, p_1 equals final pressure, and v_1 equals final volume. Initial pressure times the 1.3 power of the initial volume is equal to the final pressure times the 1.3 power of the final volume, p and p_1 being absolute pressures.

To construct the adiabatic curve from the formula $pv^{1.3} = p_1v_1^{1.3}$, it is necessary to calculate ordinates for the different

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points of the stroke. This involves the use of logarithms, which are not well understood by most operating engineers, and even if understood, require more time than can generally be devoted to that work. But the following simple method using Table VI is available. In this table the pressure at the beginning of the compression stroke is taken as 1 lb. abs. The total cylinder volume is also assumed to be 1.

TABLE VI.—ADIABATIC CONSTANTS FOR AMMONIA

Parts of cylinder, volume (1)	Absolute pressure, pound per square inch (2)	Parts of cylinder, volume (1)	Absolute pressure, pound per square inch (2)
1.00	1.000	0.50	2.462
0.95	1.069	0.45	2.824
0.90	1.147	0.40	3.291
0.85	1.235	0.35	3.915
0.80	1.337	0.30	4.783
0.75	1.454	0.25	6.063
0.70	1.590	0.20	8.103
0.65	1.751	0.15	11.778
0.60	1.943	0.10	19.953
0.55	2.175	0.05	49.129

In column 1 are give the volumes for twenty positions of the piston, and in column 2 are given the corresponding pressures as calculated by the foregoing forumla. The clearance can be disregarded as being too small to appreciably effect the results to be obtained. By means of this table the absolute pressures for the given parts of the volume of any cylinder can readily be calculated by multiplying the corresponding values in column 2 by the initial absolute pressure—that is, the pressure in the cylinder when the piston starts on the compression stroke. For example, if the initial pressure is 30.7 lb. per square inch, absolute and the gas is compressed three-tenths—that is, to 70 per cent of its original volume—the value in the table opposite 0.70, or 1.590, is multiplied by 30.7, and the product 48.813 will be the pressure per square inch in the cylinder at that point.

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The following illustration will make clear the application of the table when constructing an adiabatic curve: Let *ab* Fig. 10 represent the curve of a properly operating ammonia compressor with a suction pressure of 16-lb. gage and a discharge pressure of 160-lb. gage. Let *AA* represent the atmospheric pressure line, *L* the length of the stroke and *VV* the vacuum line. Divide *L* into 10 equal parts, erect vertical

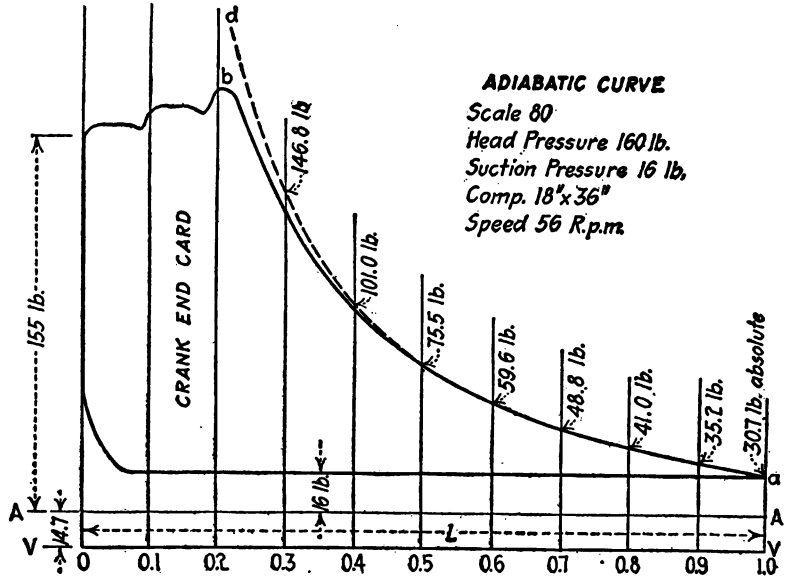


FIG. 10.—Method of constructing adiabatic curve.

lines at these division points on the line *VV*, and number the points from 0 to 1 by tenths, as shown. Then the distance *Va* will represent the suction gage pressure of 16 lb. plus the atmospheric pressure of 14.7 lb., or 30.7-lb. absolute. To get the point on the vertical line 0.9 for the curve, take the value 1.147 from column 2, Table 14, opposite the 0.9 in column 1, and multiply by the initial absolute pressure of 30.7 lb. Thus $1.147 \times 30.7 = 35.2$ lb. per square inch, nearly, which is to be laid off on the vertical line 0.9 from the vacuum line, to the same scale that *Va* represents 30.7 lb.

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Next take the point 0.8 and find the corresponding pressure, column 2, to be $1.337 \times 30.7 = 41.04$ lb. nearly; then in a like manner do the same for 0.7, 0.6, 0.5, etc. When all the points have been located, the curve *ad* may be drawn through them as shown by the dotted line *ad*, which is the adiabatic curve.

If the ammonia gas were compressed from the point *a*, Fig. 10, up to the condenser pressure in a perfectly tight and non-conducting cylinder without loss or gain of heat, then the adiabatic line would be the curve traced by the indicator

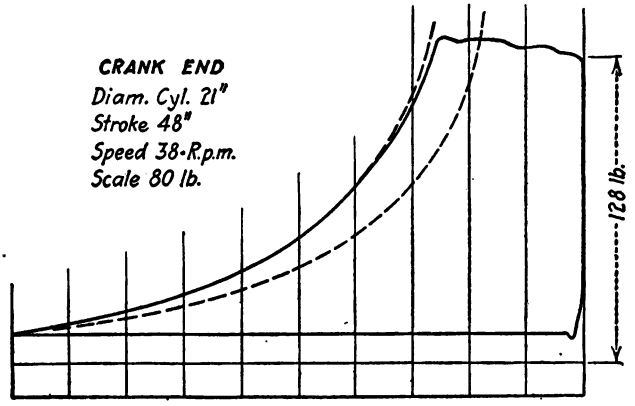


FIG. 11.—How expansion line would follow adiabatic curve if there were no leaks.

pencil. If there is no leakage past the valves or piston, the compression line will in all ammonia compressors almost exactly follow the adiabatic line, departing from it only as shown in Fig. 11.

It makes no difference whether the compressor is single-acting or double-acting, or operating on the wet or dry principle, if it is in good condition the compression curve as traced by the indicator pencil will closely follow the adiabatic line. The only operating condition that could cause the compression curve to lie below the adiabatic curve, if the compressor is in good condition with no leaky valves or piston, would be an excessive amount of liquid ammonia. In this case the com-

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pression curve will drop toward the isothermal curve. The ammonia compressor was designed to pump ammonia gas and not liquid ammonia.

Figure 11 shows the compression curve following the adiabatic curve closely, or how it would follow if the compressor were in first-class condition. The compressor from which Fig. 10 was taken is also in first-class condition.

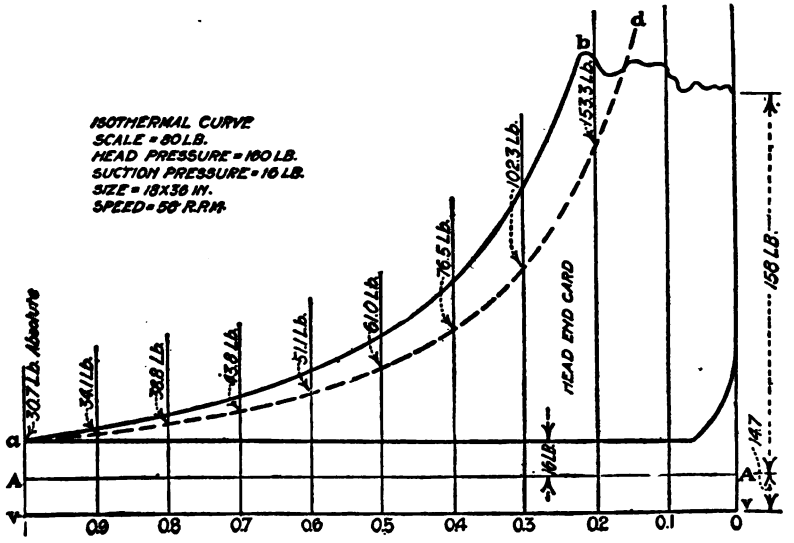


FIG. 12.—Constructing isothermal curve.

If the heat of compression could be removed as fast as generated—that is, if the temperature could be kept constant—then the indicator would trace the isothermal line and there would be a considerable saving of work, but this cannot be done; the temperature should, however, be kept as low as possible by the proper circulation of sufficient jacket water.

The isothermal curve is more easily calculated than the adiabatic curve by using the formula $pv = p_1v_1$, which is the law of isothermal expansion or compression; that is, the initial pressure times the initial volume is equal to the final pressure times the final volume. The diagram is divided and vertical division lines erected the same as for the adiabatic line.

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Assume the point a, Fig. 12, as the starting point. At this point, $p = 30.7$ lb. and $v = 1$. When the volume is reduced to 0.9 of the original volume, or when $v_1 = 0.9v = 0.9 \times 1 = 0.9$, the corresponding pressure is found from the formula $pv = p_1v_1$. Substituting the values of $p = 30.7$, $v = 1$ and $v_1 = 0.9$, the formula reduces to $30.7 \times 1 = 0.9 \times p_1$, from which $p_1 = \frac{30.7}{0.9} = 34.1$ lb. Now lay off this pressure on the vertical line 0.9 and proceed in the same manner for 0.8, 0.7, 0.6, etc. Figure 8 shows a diagram where this is done. Thus, when

$$
\begin{aligned}
v_1 &= 0.8p_1 = 30.7 \div 0.8 = 38.8 \text{ lb.} \\
v_1 &= 0.7p_1 = 30.7 \div 0.7 = 43.8 \\
v_1 &= 0.6p_1 = 30.7 \div 0.6 = 51.1 \\
v_1 &= 0.5p_1 = 30.7 \div 0.5 = 61.0 \\
v_1 &= 0.4p_1 = 30.7 \div 0.4 = 76.5 \\
v_1 &= 0.3p_1 = 30.7 \div 0.3 = 102.3 \\
v_1 &= 0.2p_1 = 30.7 \div 0.2 = 153.3
\end{aligned}
$$

By laying off these values of p_1 as ordinates for the corresponding volumes, the points for the curve are located and a curve ad drawn through these points will be the isothermal curve.

While the adiabatic curve is the true guide for comparison of the compression curve, the isothermal curve in addition to the adiabatic gives it a more definite location and better comparison.

A compression curve may appear all right, yet when the true adiabatic and isothermal curves are drawn, it will at once become apparent whether it is defective or not.

The adiabatic curve will be taken up first (see Fig. 9). Notice that the compression line is above the adiabatic line. This indicates that the pressure in the cylinder increases faster than could be caused by the mere compression of the gas by the piston. The reason is that high-pressure gas is leaking into the cylinder through leaking discharge valves, their gaskets, or through the cylinder-heat gasket between the discharge port and the cylinder.

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It is evident that if this condition exists, the cylinder is not getting its full charge of gas and therefore not enough gas is pumped. It is evident from the increased area of the diagram that it requires considerable power to run the compressor. This means a loss, and it does not take long for a compressor to waste much fuel if allowed to run in this condition.

The valves should be removed, the cylinder head taken off and all gaskets carefully examined and, if found defective or looking bad, replaced with new ones. If the machine has no valve-cage gaskets but has ground joints, regrinding may be necessary to get a tight joint. The joint may be all right, but the valve cage may have worked loose, causing the seat to leak. The valves might leak and require regrinding. Generally, an inspection of the valve seats and disks will show if any are leaking. One will generally find the valve seat and disk pitted, caused by scale from the pipe system. Pieces of this scale will lodge on the valve seat and the continual closing of the valve on the small hard pieces will cause the seat and disk to become pitted with numerous small holes. This condition may not exist, but a careful examination of the valve seat will show whether the bearing surface is good or bad. It may be possible that there is a poor bearing surface caused by the valve disk not seating true. Whatever the cause, the valve should be put in good condition and reground if necessary. After regrinding, it can be tested for tightness by placing on a table in an upright position and filling it full of gasoline, letting it stand for a few hours. If it will hold gasoline it is tight.

All compressors should have an extra set of valves kept in first-class condition, so that any defective valve can be replaced in a short time. Do everything possible to remedy the trouble and then take more diagrams. Draw the adiabatic line, and if the compression curve is still above the adiabatic, you have not remedied the trouble.

Referring to Fig. 13, it is seen that the compression curve falls away rapidly toward the isothermal curve, and the reason is that the pressure is not as great as it should be at any point on the curve. The cause for this is that some of the gas has

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leaked out of the cylinder; the piston, the suction valves, some gasket or joint leaks, which causes part of the gas to be pumped over and over again, thereby reducing the capacity of the machine. It frequently happens that all of these defects exist at the same time.

A careful examination of all gaskets and joints should be made. If it is found that the valves leak, they should be ground in, the cylinder should be calipered to see if it is truly

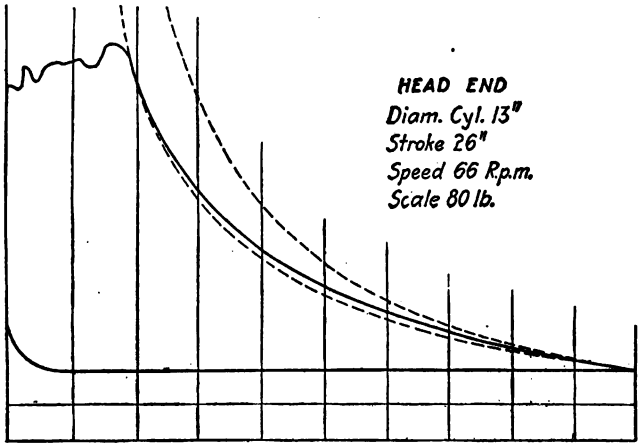


FIG. 13.—Compressor has leaky valves or pistons.

round, and if found out of round, badly cut or scored, it should be rebored and new piston rings fitted.

The piston and rings should also be carefully examined. A new set of piston rings might be necessary, and they should be a good fit in the grooves of the piston. If the piston rings are set out with springs, be sure that they are all right and in the proper position. Examine the valves and cylinder carefully, after which the compressor should be indicated to be sure the trouble has been corrected. In the absence of an indicator the pressure gage may be used by attaching it to the indicator connection; then compress the gas in the cylinder to a high pressure and the reading on the gage. If it does not decrease or decreases very slowly, the trouble has been corrected. If it decreases rapidly, another examination should be made.

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In making an analysis of an indicator diagram from an ammonia compressor, any deductions are apt to be misleading unless one is familiar with all conditions at the time they were taken; for, as stated in the preceding pages, conditions are influenced by many factors.

Aside from the foregoing, experience has taught that it is necessary to have an intimate knowledge of the construction of the compressor from which the diagrams were taken and a knowledge of the indicator used and the engineer who used it.

The careful operator will be sure to wipe clean and dry the indicator and its parts and finally wipe them with an oily cloth before putting them away.

It is well known that many defects of ammonia compressors can be easily detected without the use of an indicator; on the other hand, it is also a fact that many defects cannot be so readily detected and are brought to light only by its use. Further, experience has proved to me that 90 per cent of the ammonia compressors in operation will show some defective condition; in fact, you will find few compressors that will give a diagram like that illustrated in Fig. 11, which indicates that the compressor was in first-class condition and properly operated.

It is sometimes confusing to the inexperienced engineer to read diagrams from defective compressors; also experienced engineers sometimes have considerable trouble in locating defects that are brought to light by the indicator. For these reasons numerous diagrams are illustrated showing defects that are most commonly met. A careful examination of these will assist one to make a comparison with his own diagrams. All diagrams illustrated are reproductions from actual ones and represent all kinds and makes of machines operating under all conceivable conditions.

Figure 14 represents a card taken from a machine with a cut or scored cylinder. Notice how the compression curve falls away from the adiabatic curve and toward the isothermal curve, and how large is the reexpansion loss. The suction valves were also found to be in bad condition; in fact, the whole machine was in a deplorable state. This compressor was

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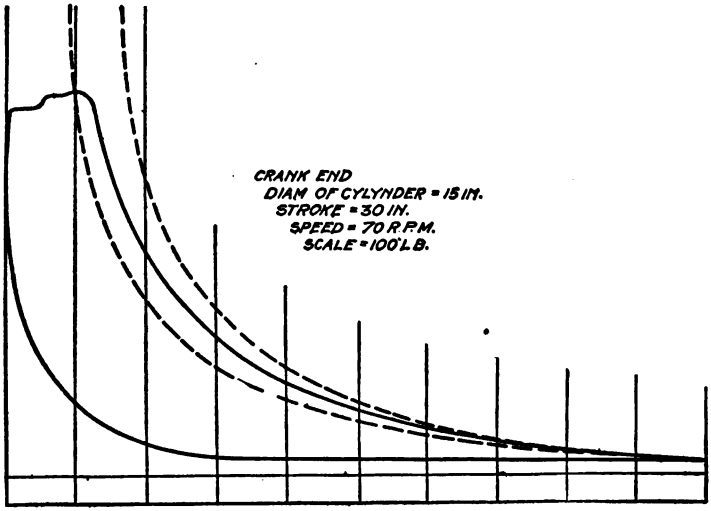


FIG. 14.—Compressor has a cut or scored.

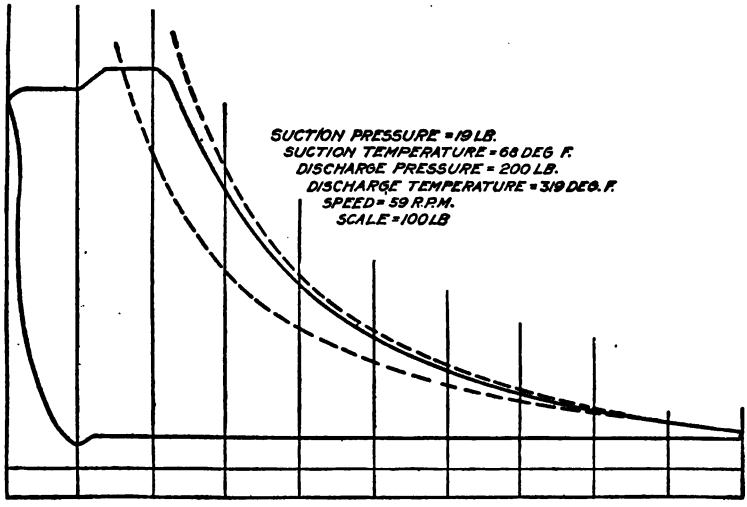


FIG. 15.—Compressor with leaky discharge.

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operated in this condition for a long time before it was suspected that any defect existed, and it is needless to say that the capacity of the machine was much below normal, thereby causing a considerable loss. The plant where this machine was located had an indicator, but it was never used.

Figure 15 shows a peculiar condition of a discharge valve. High-pressure gas was leaking back into the cylinder through the valve. Ordinarily when this condition exists the compression lines lies above the adiabatic curve, but in this case it

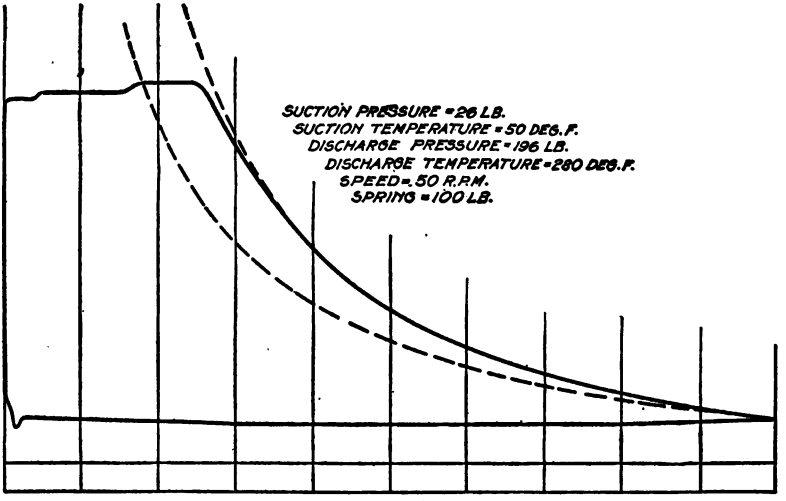


FIG. 16.—Fig. 15 after valve and cage were made tight.

falls below it, the reason being that the gas during compression leaked out of the cylinder from the same source that allowed it to leak back in after discharging.

Considerable trouble was had before the cause of the trouble was located, as the valve and cage seat were apparently all right. It is customary at this plant to change valves often, thereby always having the compressor valves in first-class condition, but in making this particular valve change not quite enough attention was given to ascertain if the cage seat was true and had a good bearing; it was later shown that it did not, with the result that when the valve and its cage were

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securely tightened in the housing, the seat was sprung, which caused the leakage. Upon applying the indicator after remedying the trouble, Fig. 16 was obtained.

Cases of this kind show the value of the indicator. This particular trouble is one of many that cannot be detected without its use. While it is true that the leak was not serious, still it was a leak, and leaks reduce efficiency.

Figure 17 shows a diagram from a compressor that had a broken discharge-valve stem. The writer was present when

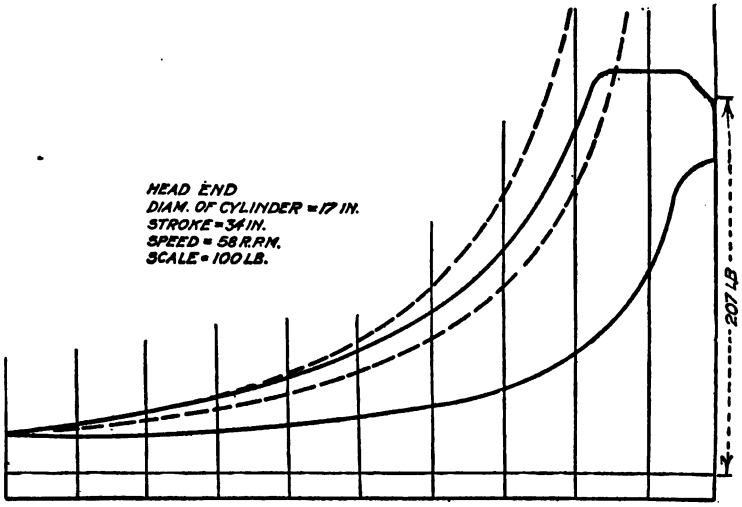


FIG. 17.—Compressor unit, broken discharge valve stem.

the compressor was overhauled, and after it had been in operation a few days, diagrams showed it to be in good condition.

A few days later, while indicating other compressors in the same plant, the indicator was again applied to this same compressor, nothing wrong being suspected or noticed in its operation, when the card, Fig. 17 was obtained. This is a striking illustration of the value of indicating the compressor at regular intervals.

Some engineers might argue that a broken valve stem in a compressor would make itself known to the operator immedi-

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ately. Sometimes it will and sometimes it will not—it all depends upon the nature of the break, valve construction, etc. Take a plant with only a single compressor and working to the limit of its capacity. Any falling off in capacity would be investigated immediately, and perhaps a defective compressor brought to light. Now suppose the compressor was much too large for the work it was doing. With no apparent reduction in output, or in a plant where there were several machines operating, it would not be so readily noticed.

Figure 18 shows a diagram from a compressor with a suction-valve spring that is too stiff. This trouble may also be

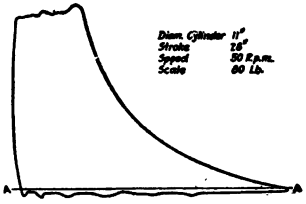


FIG. 18.—Suction valve spring too stiff.

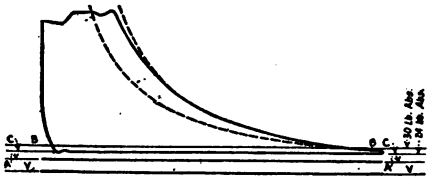


FIG. 19.—Suction valve port too small.

caused by the valve stem binding in its guide or valve cap, or by too small valves and ports for the admission of the gas. The line *AA* shows the suction gage pressure to scale as indicated by the gage. Sometimes the suction-pressure line will be smooth, as in Fig. 19, also from the foregoing causes, but more generally the smooth straight line is caused by too small suction-valve area, port area or small suction pipes, excessive friction of gas in the suction pipes or in the refrigerating coils. The part of the diagram between lines *B* and *C* represents a loss of compressor efficiency because the machine is not filling with gas at the gage pressure. If the port or valve area, suction-pipe area, or excessive friction, etc., is the cause, speeding up the machine will reduce the efficiency still more; if port and pipe areas are ample and no excessive friction exists, the trouble is in the valve spring or the valve itself.

Figure 20 shows a diagram from a compressor in which the discharge pressure is too high. This may be produced by

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small discharge valves, small pipe area or by too stiff valve springs. More power is required than would be necessary if the valve springs were not so stiff.

Figure 21 is from a compressor that has a suction valve that sticks or works sluggishly. This causes the pressure



FIG. 20.—Restricted discharge passage.

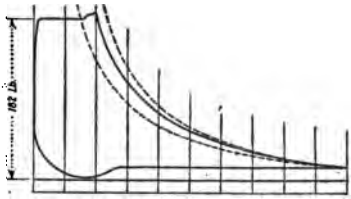


FIG. 21.—Suction valve sticks.

in the cylinder to drop too low at the beginning of the suction stroke.

Figure 22 shows another form of sticking suction valve.

Figure 23 is from a compressor of the vertical single-acting type. This is from a small plant having but the one machine, and in midsummer a reduction of capacity was noticed. The valves were exchanged, a reserve set being on hand, with no

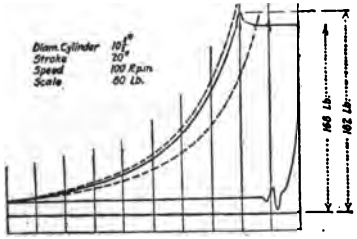


FIG. 22.—Another sticking suction valve.

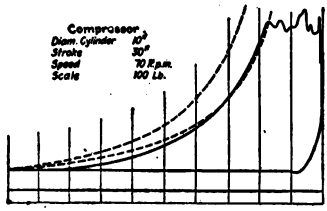


FIG. 23.—Leaky head gasket.

results. The cylinder head was then removed, the cylinder and piston examined, but nothing found wrong. The machine was assembled, and when the valves were again examined no defect could be found in them.

The cylinder head was again removed and the gasket examined and nothing wrong noticed. The gasket was

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removed, and after closer examination it was found that there was a split in the gasket about one inch long, directly over the partition that separated the suction and discharge ports. This split was closed so neatly that if the gasket had not been removed it would not have been discovered—at least not so soon. A new gasket remedied the trouble. This incident

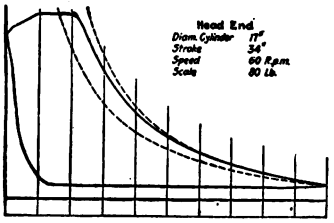


FIG. 24.—Leaky discharge valve cage.

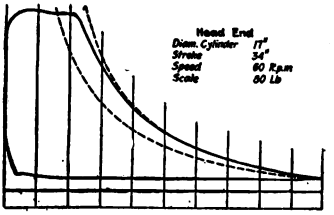


FIG. 25.—After cage was readjusted.

illustrates how deceiving things are sometimes and how careful one must be in looking for defects in compressors.

Figure 24 is from a horizontal double-acting compressor where the valve-cage seats in the cylinder heads are a ground joint; one of these was leaking. After grinding the cage to a good seat, Fig. 25 was obtained.

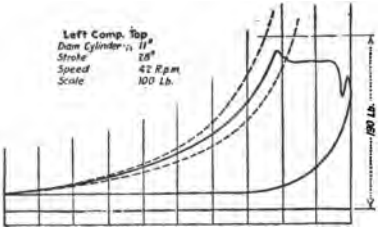


FIG. 26.—Sticky discharge valve.

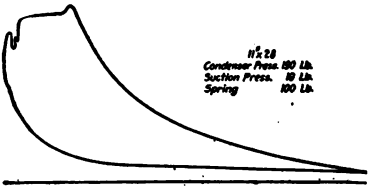


FIG. 27.—Insufficient discharge valve cushion.

Figure 26 was taken from a vertical double-acting compressor. Notice the hook caused by the discharge valve sticking when closing. Nothing was suspected to be wrong with this machine, and the trouble was only revealed by application of the indicator. Another valve was substituted, and a few days later a better diagram was obtained. Upon examination the

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trouble was found to be lack of oil. The defective valve action was in the upper heads, and the upper and lower heads were connected to the suction pipe in such a manner that the lower end valves got entrained liquid or oil passing along with the gas, while the valves in the upper head got only the driest gas. The valves, upon removal and examination, were found to be dry, and a dry red dust was found in them. A lubricator fitted to upper suction bend eliminated all danger of more trouble from this source. Figure 27 is a diagram almost identical with Fig. 26 but was actually caused by too little

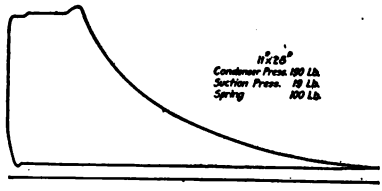


FIG. 28.—Diagram after drilling second cushion hole.

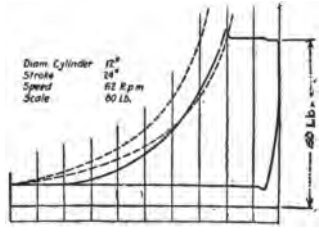


FIG. 29.—Incorrect cushion arrangement.

cushion for the discharge valve. By drilling a second hole in the valve cushion cap the diagram in Fig. 28 was secured. In case where the hole in the cushion cap was improperly placed the compressor gave the diagram in Fig. 29.

The diagram in Fig. 30 was from one end of a horizontal double-acting compressor, having a binding discharge valve. The suction valve is not tight enough to resist the high-pressure gas toward the later part of the stroke, and some of the compressed gas leaks back into the suction pipe. The discharge valve also sticks or leaks and allows some of the discharged compressed gas to leak back into the compressor, causing a large reexpansion loss before the suction gas enters. The compressor from which Fig. 30 was taken was much too large for the work it was doing, being of 50-ton ice-making capacity and working on a 25-ton freezing tank and doing no storage service. The effects of the leakage were therefore not noticed and were only brought to light when the indicator

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was applied. After the valves and cages were ground in tight, the compressor did the same work on 12 revolutions less speed, thereby saving much steam and coal.

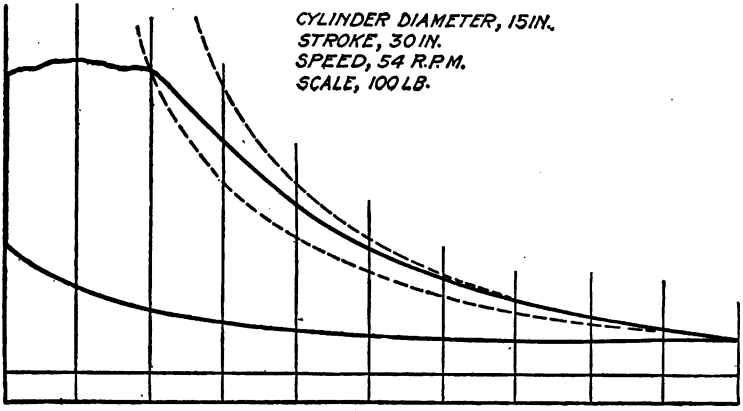


FIG. 30.—Leaky suction valve.

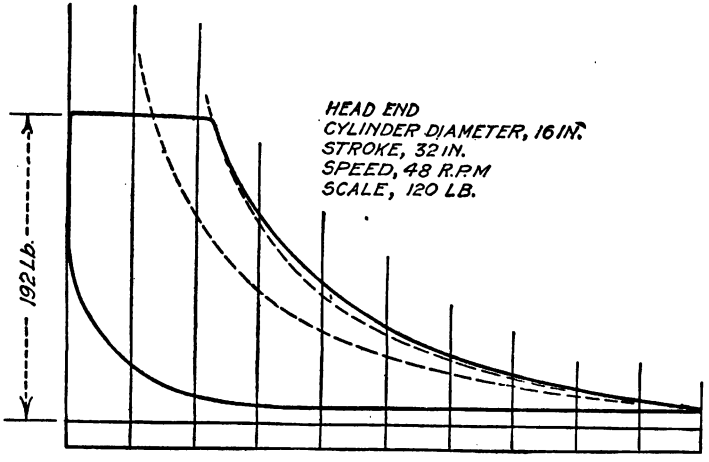


FIG. 31.—Leaky discharge.

Figures 31 and 32 illustrate another instance where it was not thought necessary to indicate the compressors. There are several machines at the plant where these diagrams were taken, and as long as the work was done it made no difference

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how many machines were run to do it or at what speed they operated. The machine from which these diagrams were taken might as well have been shut down as far as the amount of work it was doing was concerned. The head-end diagram, Fig. 31, shows leakage of high-pressure gas back into the cylinder after having been discharged into the discharge pipe, caused by either a leaky discharge valve or discharge-valve gasket, or cylinder-head gasket between the cylinder and discharge port. This leakage of the discharge valve also

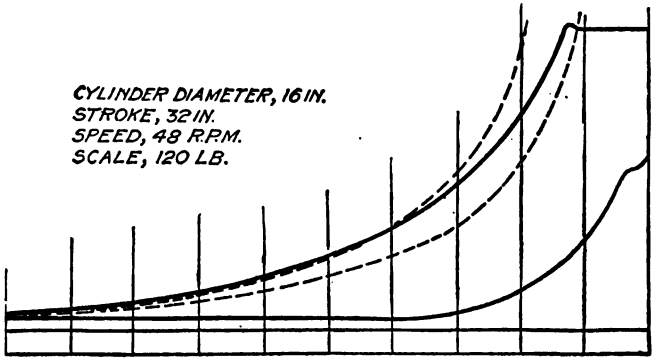


FIG. 32.—Leaky piston.

causes the large reexpansion loss shown. The crank-end diagram, Fig. 32, indicates a leaky piston, as shown by the rapid falling off of the compression curve at the upper part of that line and also by the abrupt hook in the reexpansion curve. The losses due to reexpansion on these curves are from 30 to 35 per cent. The suction-gage pressure was 20 lb. at the time the diagrams were taken, but the suction pressure in the cylinder, as scaled on the cards, is from 9 to 10 lb. This drop of 10 lb. means considerable loss of capacity and is due to the suction pipe, suction valve and port are being too small. This plant had a good engineer, who indicated the steam engines regularly, but he never thought it was necessary to indicate the compressors. There are many engineers of the same opinion. The facts disclosed by Figs. 31 and 32 admit

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of no argument as to the value of indicating the ammonia compressor. It sometimes happens that some foreign substance finds its way into the valve mechanism and can cause trouble. Such an occurrence is shown in Fig. 33. An examination revealed a piece of lead gasket in the spring, which retarded the opening and closing of the valve. Notice

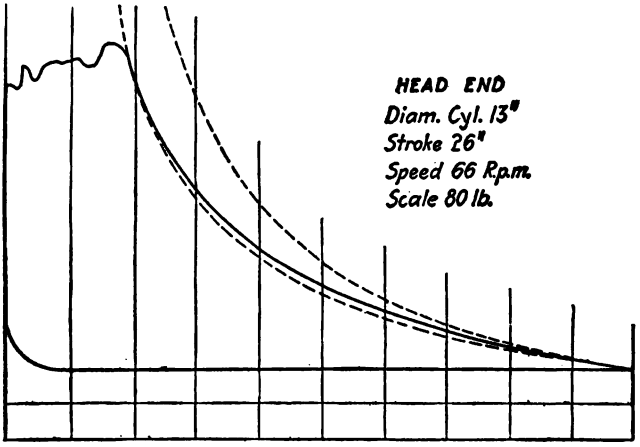


FIG. 33.—Defective discharge valve.

the reexpansion curve; notice also that the line of suction pressure gradually drops to nearly the atmospheric line.

INDICATING THE SUCTION AND DISCHARGE LINES

It is of interest to take diagrams from the suction and discharge pipes at the time the compressor is indicated. A special fitting is used, which connects the compressor cylinder to the indicator and to the suction and discharge pipes. The advantage of this is that all diagrams may be obtained without changing the indicator around to the different locations. Figure 34 shows one layout of the connection.

It is important that there be no lost motion in the indicator for pipe diagrams and that fine, distinct lines be traced, because of the close proximity of the lines. Study the conditions as revealed by these diagrams and compare them with the

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compressor-cylinder conditions. The pressure gages may be checked by the use of the indicator.

In taking suction-pipe diagrams use a 16- or 20-lb. spring or any low-pressure spring that closely corresponds to the suction pressure carried, instead of the high-pressure spring

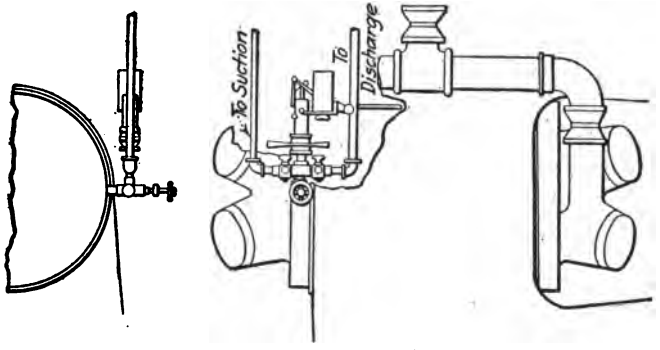


FIG. 34.—Suction and discharge indicator connections.

used for cylinder or discharge-pipe diagrams, as the low-pressure spring will show up variations in pressures impossible for the high-tension springs to reveal on such low pressure.

Figure 35 shows a diagram from another compressor with discharge-pipe pressure diagram and also the suction-pressure diagram. The compressor and discharge-pipe diagrams were taken with an 85-lb. spring and the suction-pipe diagram with a 20-lb. spring. Observe in this case that the suction-pressure diagram shows no variation of pressure in the suction pipe when the piston moves forward from the end of the storke. Notice also the increase of the pressure on the diagram from the discharge pipe and its relation to the discharge gas from the compressor, as shown by the diagram.

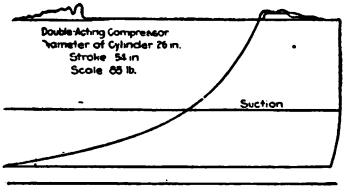


FIG. 35.—Suction discharge and cylinder diagram.

It seems to be the opinion of most engineers that the reduction of clearance in the compressor to the lowest limit is of the

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greatest importance; but the consequence of this is much overrated. Clearance space has only a slight effect on the efficiency of a compressor, but it is detrimental to the capacity, and the greater the difference in the pressures the greater the losses. The work per unit of volume pumped is practically

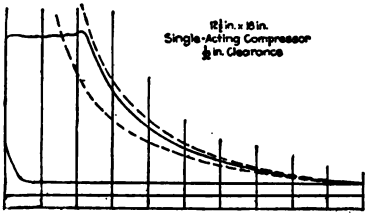


FIG. 36.

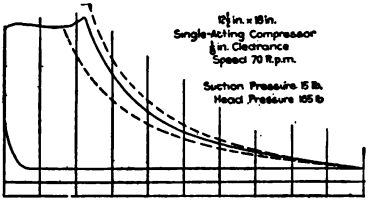


FIG. 37.

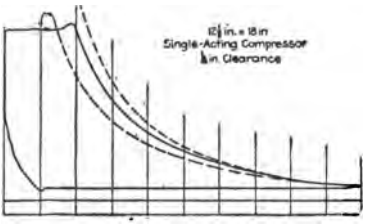


FIG. 38.

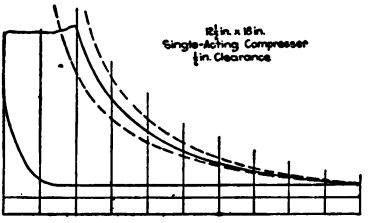


FIG. 39.

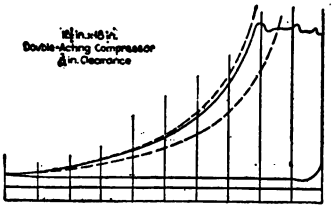


FIG. 40.

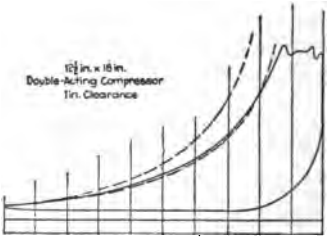


FIG. 41.

FIGS. 36–41.—Testing clearance losses.

the same, zero clearance or not. That is, assume a compressor with zero clearance, then give it clearance; the capacity is reduced, but the amount of work done per unit of capacity, other things remaining equal, is practically unaltered.

In double-acting compressors the clearance is fixed and cannot be altered much. In single-acting compressors it can be

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reduced to the lowest mechanical limit, and this offers an advantage over the double-acting machine. There is generally more clearance in a compressor when in operation than is supposed, as loose bearings can produce excessive clearance, and the amount of re-expansion which follows can best be determined by the indicator.

The diagrams, Figs. 36 to 41, are from tests taken expressly to determine the losses due to clearance, by one of the largest compressor builders in this country. Each states the type of machine from which it was taken—that is, single- or double-acting—also the linear clearance space.

Some interesting deductions were made from these clearance tests relative to the effect on horsepower required per ton, the effect of clearance on the capacity of compressors and the relative losses due to clearance and re-expansion of single- and double-acting machines.

Table VI will show the effect of capacity and horsepower per ton due to clearance, based on 15.67-lb. suction pressure, 185-lb. condenser pressure and 70 r.p.m.

TABLE VI.—EFFECT OF CAPACITY AND HORSEPOWER DUE TO CLEARANCE

Linear clearance, inch	Clearance volume per cent of displacement		Single-acting 15.67 lb. suction pressure		Double-acting 15.67 lb. Suction Pressure	
	Single-acting	Double-acting	Tons refg.	Compressor i. hp.	Tons refg.	Compressor i. hp.
1/32	0.24	....	38.0	1.30	....	....
3/64	....	0.42	....	....	33.0	1.60
1/8	0.76	0.85	37.2	1.32	21.1	1.62
1/4	1.46	1.55	35.6	1.34	30.0	1.64
1/2	2.85	2.93	34.4	1.36	28.9	1.72
1	5.63	5.71	29.7	1.39	22.9	2.01

Figures 42 to 44 show small clearance from both single- and double-acting compressors and are from small and large compressors. These machines have never given any trouble operating with such small clearance, and unless very carefully operated and handled, I would not advise reducing the clear-

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ance to such a point. It is much safer and better to allow a little more clearance, and it would cause no serious loss if the compressor is not handling liquid (which it ought not to), as liquid would cause a re-expansion loss if the machine had too much linear clearance. Even with small clearance liquid would cause re-expansion losses, much more so than a compressor with greater linear clearance handling only dry or saturated gas; and besides, the presence of liquid in the cylinder is apt to cause damage as well, as loss in efficiency.

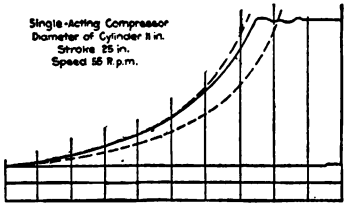


FIG. 42.

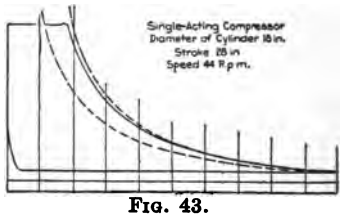


FIG. 43.

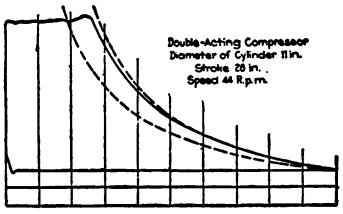


FIG. 44.

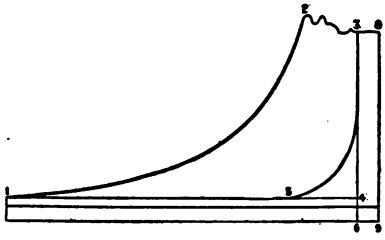


FIG. 45.

FIGS. 42–44.—Diagrams showing small clearances.
FIG. 45.—Clearance diagrams.

Referring to Fig. 45, lines 1, 2, 3, 4, 5 show the clearance diagram, line 1 to 4 represents the compressor stroke and line 3 to 8 represents the clearance volume, including linear clearance and that of the passages and ports. The gas is discharged at pressure p_2 along the line 2-3, and at the point 3 the piston reverses and the discharge valve closes. The volume of gas represented by the clearance space 3-8 expands along 3-5 until at 5 the suction pressure is reached and the suction valve opens. The volume drawn from the suction pipe is now 1-5 and not that of the piston displacement 1-4;

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the ratio of 1-5 to 1-4 is called the apparent volumetric efficiency.

The apparent volumetric efficiency becomes the real volumetric efficiency when the temperature of the gas, after it has entered the cylinder, is equal to the temperature of the outside gas. During admission to the cylinder the temperature of the gas is raised by coming in contact with hot surfaces, which superheat the suction gas; thus the weight of gas admitted will be decreased in the ratio of absolute temperatures before and after admission. If T_1 is the suction temperature of the gas, and T_2 the gas temperature in the cylinder at the completion of the suction stroke, the expresison for volumetric efficiency must be the apparent volumetric efficiency multiplied by $\frac{T_1}{T_2}$. This loss the indicator does not show.

Due consideration must also be given the inlet passage, which may be too small for the volume, so that the gas at the completion of the suction stroke may have a lower pressure in the cylinder than in the suction pipe; on the other hand, it is also possible, by means of large ports and suction pipes, to have the pressure at completion of the suction stroke greater than that in the suction pipe, because of inertia of the gas in the long pipes. The piston reversal suddenly checks the gas flow and may cause the pressure to rise in the cylinder.

Most ammonia compressors leave the shop with holes drilled and tapped for applying an indicator generally in the side of the cylinder. Into the hole so prepared screw the indicator cock directly whenever possible. If it is necessary to use any pipe so as to clear any obstruction to get a straight line to the crosshead, use double-extra-heavy pipe. This has a smaller internal area than the ordinary ½-in. pipe. The reason for using as small a bore as possible is to reduce the clearance.

For indicating compressors, the arrangement, Fig. 46, should not be used either in a horizontal or a vertical compressor cylinder. It cannot be depended upon for accuracy, as it produces distortions by wiredrawing and clearance.

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Figure 47 is still worse, where angle valves are used at the ends of the cylinders and connected with side pipes in the center of which is a tee for the indicator cock. This arrangement causes still more clearance space than that shown in Fig. 52, and besides, the angle valves might leak. The foregoing applies also to the vertical type of ammonia compressor.

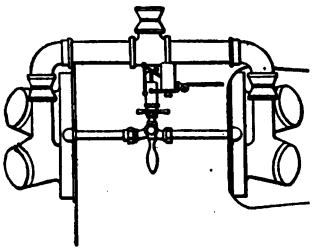


FIG. 46.—Improper indicator connections.

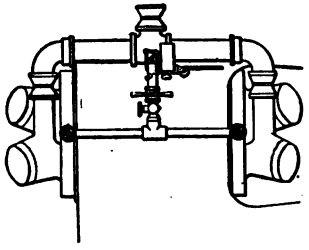


FIG. 47.—Another poor arrangement.

In putting up piping or connections for use with the indicator, use no red lead, litharge or other mixture, as it is likely to cause trouble. Good sharp, clean-cut threads with a little oil are usually all that is required. Figure 48 shows the result of indicator connections made up with litharge in a short pipe

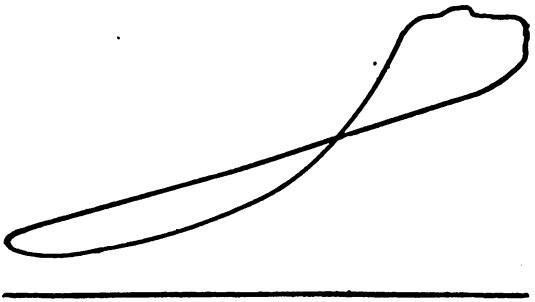


FIG. 48.—Effect of litharge on pipe connections.

connection between the stationary indicator cock and compressor cylinder. When the pipe was screwed tight, the litharge was forced ahead in a loose lump and hardened, and when the indicator was applied this lump of litharge acted as a check valve, producing the diagram shown.

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Every possible precaution should be taken to remove from the indicator connections all scale, dirt, etc., that would be liable to get into the indicator cylinder. Blow all pipe and connections thoroughly before attaching the indicator, a little particle of dirt can do a great deal of damage to the cylinder. Besides, dirt will also cause some freak diagrams, such as shown in Fig. 49, taken from a compressor that had an accumulation of dirt in the indicator-pipe connections. After this pipe was thoroughly cleaned, a good diagram was taken.

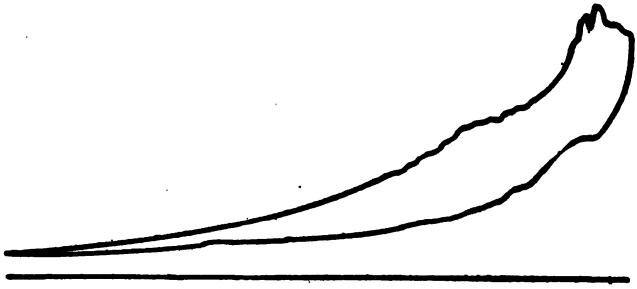


FIG. 49.—Diagram with constructed connections.

Before and after using the indicator, clean it thoroughly and oil it carefully. The indicator is like the engine, only more delicate and sensitive, and if one expects it to work freely and smoothly, it must be carefully cared for. Be sure there is no lost motion. This is particularly to be guarded against in indicators, and few are free from this defect. Freedom from friction is important.

The springs are the vital part of the indicator, and the value of all work depends upon their correctness. Be careful to select the spring with the proper tension adapted to the pressures and speeds that will be encountered. By using springs that are too weak or too strong, a distorted diagram will result. This effect is particularly noticeable in ammonia-compressor diagrams, because there are many factors that influence the diagrams and often make them misleading, and by using springs unsuitable for the pressures, you are multiplying the distortions that are likely to be encountered.

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Figure 50 shows the effects of too stiff a spring. The condenser pressure on the gage was 176-lb., and the gage was correct. The spring used was 150-lb. As scaled on the diagram, the condenser pressure is only 143-lb., or a difference of 33 lb. Then, besides, this stiff spring gives a diagram that is much too small, and it is not possible to reveal defects that might exist or to make a proper analysis of the diagram. A spring that is too weak for the pressure or speed encountered will distort the diagram. Different types of indicators will cause distortions in different ways, owing to the use of springs that are too stiff or too weak for the pressures and speed.

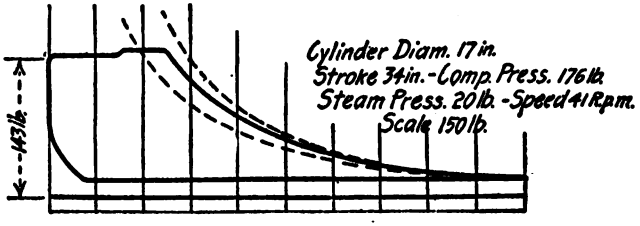


FIG. 50.—Stiff indicator spring.

The error in the final results will be less if the diagram has a large area, and the smaller the area of the diagram the larger will be the error in the final results.

Another important item in taking diagrams is the pencil point. Do not use soft lead; it makes too wide a line and is not clear—it is more or less blurred. Use a hard lead with a sharp point. If you use metallic points, keep them sharp-pointed. The pencil should trace a fine, clear, distinct line. When you take diagrams, the greatest care should be used in not pressing the pencil point too hard on the paper. There are more poor diagrams from this cause than from any other.

Use for the reducing motion cord that will not stretch, and if it is necessary to use cord carrying pulleys to clear any obstructions, make all turns or bends right angles or square and have the cord connection to the crosshead in a straight level line; that is, allow up or down no pitch of the cord. Be careful that the drum travel and reducing motion are correct

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for the stroke of the compressor. These precautions are all necessary if you wish to obtain diagrams that are correct.

An indicator built entirely of steel or aluminum is preferred for ammonia work. However, if your indicator is of brass or bronze composition, it can be used without danger of damaging the instrument if it is thoroughly cleaned and oiled afterward.

OPERATING PRESSURES

Certain applications of refrigeration require low temperatures, the production of which adds to the difficulties and the operating problems of the engineer. These low temperatures are found in cold-storage work, possibly ice-cream hardening, in oil refining and in special chemical industries of various sorts. In these applications of refrigeration, temperatures of zero deg. F. or lower are maintained, thus requiring − 15 deg. F. or lower boiling temperature of the ammonia.

At low boiling temperatures the ammonia compressor of the standard design is not efficient except at very low head pressures. At normal condenser pressures the standard machine has the following capacities per ton 20-lb. suction and 175-lb. head pressure taken as unity: Suction pressure, 40 lb., 1.611 tons; 30 lb., 1.306 tons; 20 lb., 1 ton; 10 lb., 0.6993 ton; 0 lb., 0.4101 ton; −5 lb., 0.263 ton.

From this it is seen that at low suction pressures the standard compressor fails as a medium for securing refrigerating duty, and the engineer sees clearly why the manufacturers designate the operating conditions; namely 175-lb. head pressure and 15.2-lb. suction gage pressure. In the figures given a 100-ton machine has only 26.3 tons capacity at −5-lb. suction pressure.

The reasons for lowered capacity with decrease of suction pressure have already been brought out; but it is worth while making mention of them again. These reasons are on account of the low density of the ammonia at low pressures, the small volumetric efficiency obtainable, and finally, the trouble in the condenser by overheating the ammonia.

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Some operating engineers have no clear idea of the so-called density and its effect on the tonnage of the machine. Yet in the use of steam they see clearly the reason for the relative increase in size of the exhaust-steam pipe as compared with the inlet boiler steam, especially when a low vacuum is carried as is true with the turbine operating condensing. In the steam engine or turbine the same weight of steam is exhausted as enters from the steam header, but the pressure is much lower. We say under these conditions that the *volume* of a unit weight of steam (1 lb.) is much greater with the lowered pressure. It is like the bicycle or auto air pump which compresses the air in the cylinder to a lower volume, but which has the same *weight* of air in the cylinder under both conditions.

The density of saturated ammonia, then, is less as the pressure is reduced, as may be seen from the following approximate values:

Gage pressure, pound	Cubic feet per pound
20	8.0
15	9.3
10	11.0
5	12.9
0	17.9
5	24.1

The ammonia compressor, which has a fixed piston displacement per stroke, will pump the same number of cubic feet of ammonia per minute, but will have fewer and fewer pounds of ammonia discharged out of the compressor as the pressure is lowered. And the tonnage is always proportional to the amount of ammonia *in pounds* which is pumped into the condenser per unit of time, usually taken in minutes.

The volumetric efficiency has to be considered because of the reexpansion of the gas in the clearance space at the beginning of the suction stroke. This reexpansion prevents the inlet valve from opening until the gas in the clearance volume has been reduced at least to the suction pressure. Under conditions of operation prevailing in low-temperature operation, this reexpansion factor looms up in large proportions and materially affects the effective stroke. Clearance reexpansion

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is not a direct cause of increasing the power required to compress and pump a given amount of gas, but it cuts down the capacity of the machine, as the cylinder is supplied by much less than the full cylinder volume of gas per stroke.

COMPOUND COMPRESSION

The chief objectional feature of single-stage compression as applied to ammonia is in the high temperature of discharge of the gas. In air compression the rule is usually to pump from atmosphere to about 70 lb. in the first stage and from 70 lb. in the first stage and from this pressure to the required pressure when moderate final pressures are required. An intercooler is placed between the stages, and the gas is cooled to within a few degrees of available cooling water before passing on to the second stage. The reason for so doing is to increase the volumetric efficiency, decrease the work of compression and prevent excessive discharge temperatures.

Excessive discharge temperatures are bad for ammonia. Considerable difference of opinion has been expressed on the subject, but it is clear from the practical experience of refrigering engineers that high discharge temperatures as well as high head pressures, tend to disintegrate the ammonia into hydrogen and nitrogen. These gases are permanent, and they collect in the condenser and, unless purged, will gradually increase the head pressure, like the manner of the increase in the condenser pressure when the air pump is stopped in the case of the steam engine or turbine. The formation of inert gases in the condenser has resulted in excessive purging and heavy losses of ammonia which invariably passes out of the condenser during the purging process.

Compound ammonia compression for low-temperature refrigeration has a number of important features. It is stage compression, with an intercooler between stages. It is possible to arrange the suction of the high pressure so that it will take care of the gas from the freezer rooms. This multiple-effect suction is growing popular. It is more advantageous to design in a manner similar to the compound steam engine, and to try

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to equalize the load due to compression between the two cylinders. Thus, instead of using a receiver pressure of 25- to 30-lb. gage, a pressure of some 45 to 55 lb. would have to be used, depending on the head pressure and suction pressure.

As we all know, compound compression requires an intercooler as well as an aftercooler (the condenser in refrigeration). In air, or ordinary gas compression, water is used in the intercooler, but this is not possible with ammonia because the temperature of saturation at 45- and 55-lb. gage is 30 deg. F. and 38 deg. F. respectively. It is evident that water may not be used for such cooling, unless the intercooler used water for the first part of the process and finished by means of some other cooling medium similar to the Bandalot cooler principle. The only convenient and natural medium for such low-temperature intercooling is expanded ammonia. This is done by means of an accumulator arrangement.

The functions of the accumulator are twofold. It will be remembered that under usual circumstances there is some 10 to to 15 per cent of ammonia vaporized at the expansion valve to cool the remaining liquid ammonia from the temperature of the liquid receiver to that of the temperature of evaporation (boiling) in the expansion coils. The vapor formed in the process cannot be used for refrigeration, but clogs up the expansion system unless removed The gas formed increases in amount in cases of extreme pressure range, as in the case where low-temperature refrigeration is required, and so the accumulator principle is used to remove the gas formed in precooling and to remove the superheat from the discharge gas from the low-pressure cylinder.

The gas discharged from the low-pressure cylinder is made to go through a water-cooled first intercooler and then through the ammonia-cooled second intercooler. The discharge gas is thereby cooled from, say, 100 or 150 deg. F. to about 35 deg. F. and some 5 per cent (perhaps) of liquid ammonia is vaporized in the process. Liquid ammonia not vaporized after passing into the second intercooler flows by gravity into the liquid precooler which is open at the top of the intercooler receivers, and the gasified ammonia passes out

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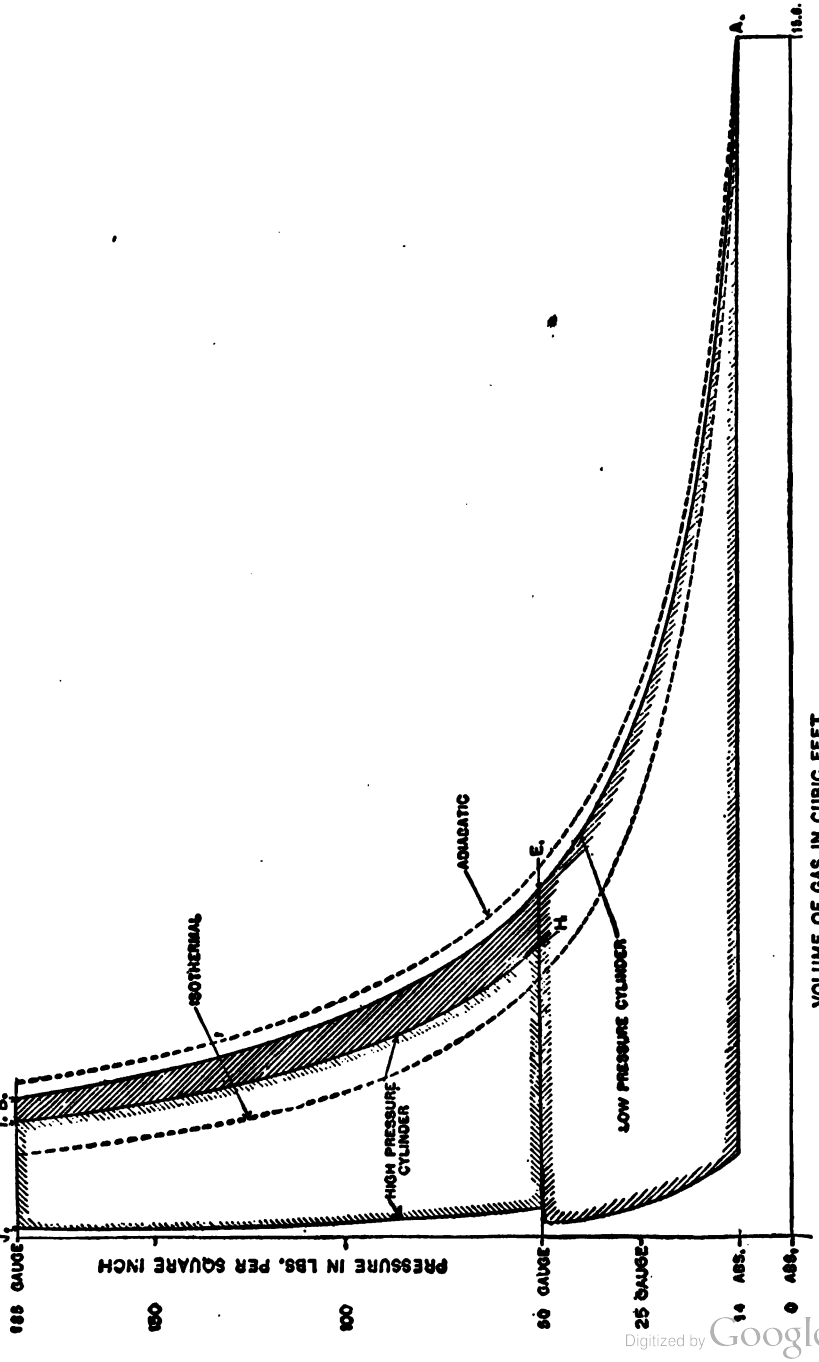


FIG. 51.—Compound compressor diagrams.

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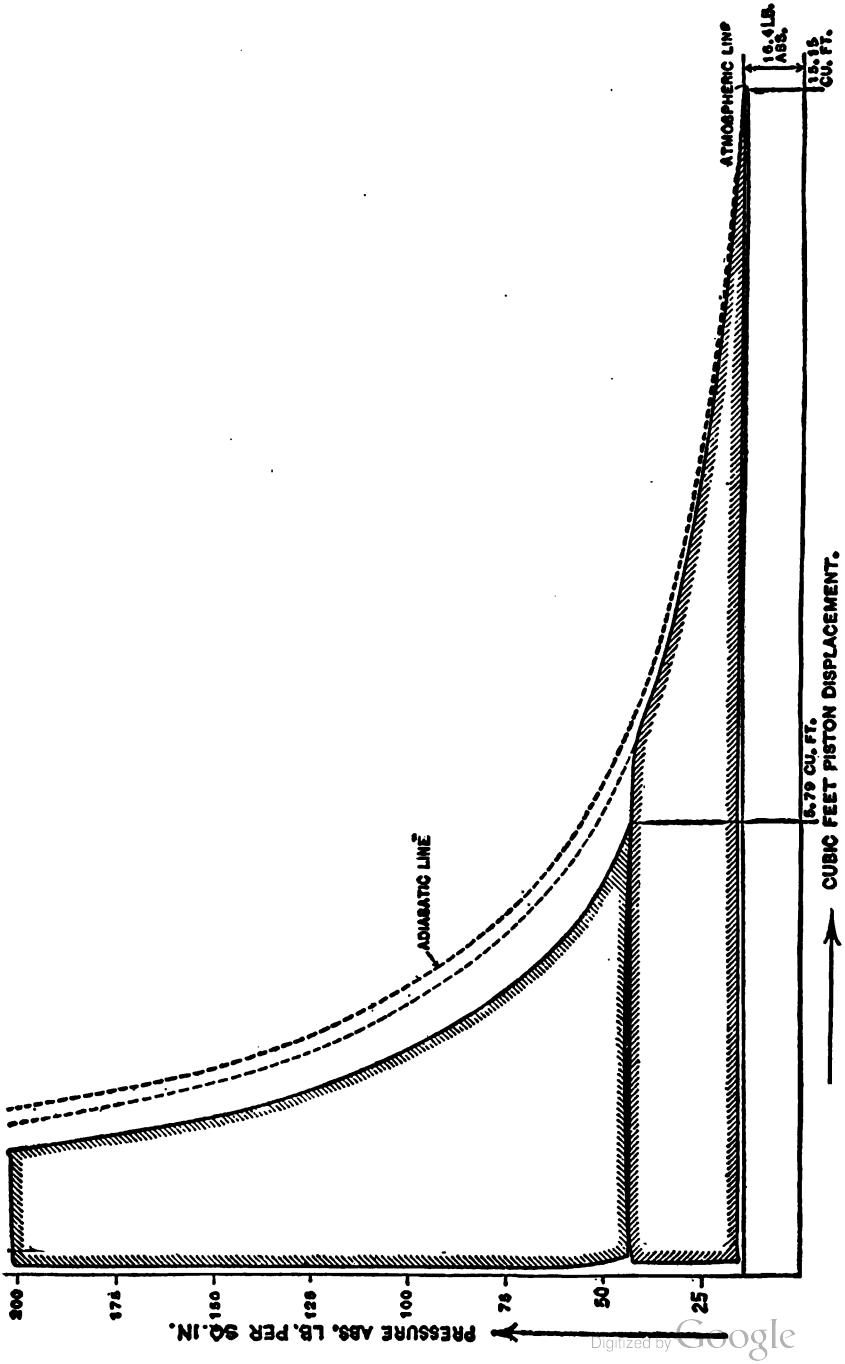


FIG. 52.—Indicator diagrams from a compound compressor.

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and enters the high-pressure cylinder of the compressor. The liquid from the liquid receivers, having passed through a long coil in contact with boiling ammonia, will be precooled to about 30 to 35 deg. F.

What is the advantage of all this? It seems that a lot of extra piping has been designed all to no purpose, as the same things occur anyway. Yet this is not true, as can be seen on careful consideration of the conditions which prevail. It is true that gas is formed by cooling the liquid precooler, but this gas is compressed only in the second stage, thus decreasing the size and the work necessary to be done in the low-pressure cylinder. Some liquid is evaporated in the second intercooler but the volume vaporized is only one-quarter of the decrease in the volume of the gas passing into and out of the intercooler. Finally, the temperature of discharge from the second-stage compressor is only 175 deg. F. (point *I*) instead of 350 deg. F. (point *B*) in Fig. 51. The clearance reexpansion loss is reduced, as seen in Fig. 52, which is taken from actual indicator diagrams.

It is seen from that the diagram that the two compressors are driven from the same shaft, which may be engine, synchronous motor, or belt driven. If desired, separate drives may be made, as is done in the Central Manufacturing Co. cold-storage warehouse, of Chicago, where large units are employed. In this plant the low-pressure cylinder is called a "booster" and liquid cooling is used in a somewhat similar manner to that already described, which is known as the D. L. Davis method. In moderate-sized plants, however, the advantage of having the two cylinders on one shaft lies in the reduced torque and less flywheel and balancing troubles.

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CHAPTER III

THE USE AND CARE OF AMMONIA COMPRESSOR

From tests that have been made it has been proven that the dry compression is the more efficient. One reason for this is that the average operator of a wet compression machine will operate his machine too wet. He either injects too much liquid at the compressor or brings the gas back from the ice tanks or coils too heavily saturated, with the result that he gets reexpansion in the cylinder which reduces the capacity of the machine. There is just as much fault in operation from bringing the gas back to the compressor superheated as supersaturated. The results are just as detrimental to the capacity of the compressor as the increase in volume due to the superheat is very large. As an example, 20 deg. superheat in the suction gas reduces capacity of compressor 5 per cent.

The influence of clearance in compressors is not so detrimental to the efficiency of a compressor, as has been claimed. It is, however, very detrimental to the capacity of the compressor and the more so the greater the difference between the high and low pressures.

From a theoretical standpoint, clearance has no influence on the work required to compress the gas, since the work performed on the gas remaining in the clearance space is again given off to the piston during the return stroke. The condition of the enclosed gas is the same at the beginning and end of the expansion, as it is at the beginning and end of the compression.

In practice, however, this is not true, because the expansion of the compressed gas is in contact with a large surface through which heat is radiated or conducted. This means that when the gas is in the compressed state it is hotter than the surrounding cylinder and heat is radiated and stored up in cylinder walls, piston and valves. When the gas first enters

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the compressor from the refrigerator it is much cooler than the surrounding surfaces and at this stage heat is conducted into the ammonia gas thereby increasing its volume and causing a loss in the capacity of the compressor. To keep the capacity uniform, the clearance should be as small as possible, or only that amount necessary to allow for expansion of piston rod when it becomes warm.

When a compressor is in operation the slack in the bearings furnish additional clearance. This is liable to be more than is generally supposed as there are three bearings that can produce excessive clearance; the pillow block, the crankpin and the crosshead pin. With loose bearings the compressor will not run smoothly and will have excessive clearance. In a double-acting compressor the clearance represents a fixed amount and cannot be less than the builders allow while in a single-acting compressor the clearance can be reduced to the lowest practical point and some claim that the single-acting compressor offers an advantage over the double-acting machine. Friction in compressors represents a loss of work and also generates heat which with the heat of compression represents a loss of from 1 to 2 per cent of the refrigeration produced. If the compressor is not properly filled with gas at the same pressure as exists in the suction line, this must be considered a loss. A reduction in pressure is mostly caused by friction of the gas through the ports and valves which not only increases the work of the engine or motor but also reduces the capacity of the compressor. The volumetric efficiency of a compressor as commonly understood is the percentage of gas discharged per stroke as compared to the amount of gas which should be discharged were there no reexpansion, and ranges from 73 to 85 per cent.

COMPRESSOR VALVES

The valves of any type or make of machine are the most sensitive parts, and the greatest of all losses in refrigerating capacity is caused by valve trouble. Such trouble may be leaks, improper spring tension sticking, breaking, derangement of parts, or restricted passages.

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For an ammonia compressor to do its full duty and so handle the proper weight of gas the valves must be gastight, of sufficient area, easy and quick in operation, and the springs must offer the least resistance possible. Scale and grit will cause the compressor valves to leak, stick and break. Small particles of scale from the inner surface of pipework, passing through the valves, will soon cut the seating surfaces, and then the gas will leak through. Pieces of scale or fine grit getting between the stem and the guide will make the valve stick, and this often causes the valve stem to break or bend.

When a new plant is built or new pipework installed, every piece of pipe and every fitting should be cleaned of scale and dirt. Every length of pipe should be stood on end and hammered to remove the scale. Fittings should be given the same treatment. When new pipework has been tested with air pressure, it should be blown out two or three times to remove loose scale. During the yearly overhauling the same precaution must be observed.

A suitable scale trap, with a fine-mesh screen must be installed in the suction line close to the compressor. This trap should be examined and cleaned at regular intervals. If this is not attended to, the passage of ammonia may be restricted or the scale may be drawn into the compressor. A few minutes shut down and a careful cleaning of the scale trap once a month may prevent a longer shut down to replace a leaking or broken valve or to rebore a cut cylinder.

The first rule for the care of ammonia-compressor valves is: Prevent scale or any foreign substance from passing through the system.

The packing of the ammonia rod on such machines as have a connection between the oil lantern and the suction line can also be the cause of valve trouble. "Chewed-up" packing will pass through this connection line and get into the valves, where it will do the same damage as scale.

High discharge temperature together with inferior lubricating oil will cause trouble, as the oil will burn on the valves, causing slow action if not sticking.

There are several things which occur that often have the

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appearance of broken or leaking valves. The most common of these are: On machines of the Boyle type the gasket under the head and between the suction or discharge chamber and the cylinder can leak, and this has the same effect as a leaking valve. Figure 53, *E* and *D*, shows places where this gasket may leak. If, as often happens, this gasket frequently blows out, the cause may be that the top of the cylinder or the head may not be faced true. If such is the case, the part that is not true must be faced. Pure gum sheet $\frac{1}{32}$ in. thick makes the best gasket for this place. When lead or other material has failed, thin pure rubber will usually give good service.

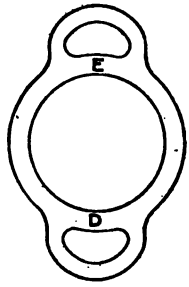


FIG. 53.—Head gasket of Boyle compressor.

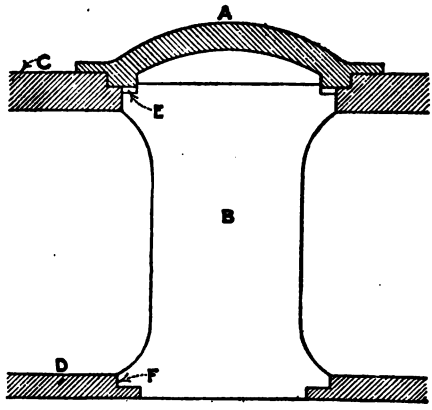


FIG. 54.—Boyle valve cage.

Leakage between the bottom of the valve cage and the head or cylinder casting produces the same effect as a leaking valve. As a rule a ground joint is used at this point, but if a gasket is required pure rubber or soft lead is best to use. A gasket cannot be used if the seating surface is not flat. On most horizontal compressors this joint is beveled and nothing but grinding with emery and oil will stop the leak.

A common cause for this kind of leak is found in such compressors where the cage is held in place by the valve bonnet. Perhaps, after grinding a cage to its seat or when putting in a new cage, the distance from the top of the cage to the

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bottom is not great enough to allow the bonnet to press the cage down hard. Figure 54 shows a short cage. To overcome this a washer or gasket of suitable thickness is cut and put in to make up for the cage shortage. Notice that this washer is narrow and does not extend out beyond the cage; *E* shows the washer, and *F* shows the joint under the cage. The washer at the top of the cage can be made of lead or rubber, but the best material is sheet iron. When the cage is of the proper length, it should be about 1/64 in. above the surrounding part of the head where the bonnet rests. A short cage moving from its seat makes a noise like a broken valve.

A sticking valve is of course the same as a badly leaking one and will sound about the same as a broken spring. A valve thought to be either sticking or broken should be taken out at once, as an accident may be the result of neglect.

Ammonia blowing through the rod packing, oil lantern and gas-relief line has the same effect as a leaking valve, as the gas passes directly from the cylinder back into the suction. Leaking pump-out connections or the valves on equalizing lines are as bad as leaking compressor valves or leaking piston.

Besides the noises accompanying short cages and sticking and broken valves, there are other similar sounds, namely: Piston loose on the rod or striking the head; too much play of the piston rings in the grooves; broken piston rings; oil lantern or packing gland working back and forth. This last often happens when the back pressure is lowered below the usual point, as in pumping down.

There is a difference of opinion as to whether small or large clearance space is harmful in an ammonia compressor. Some makers allow much clearance to insure greater safety, while others do everything to minimize it to keep down reexpansion losses. To keep the clearance as small as possible, the valves and cages are made flush with the heads. When this is done, the projection of the cage or valve through the head must be avoided.

Extra valves, cages, springs and all other parts that make the complete valve should be kept in stock. It does not pay to have parts of valves made in a local machine shop. Send to

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the factory for them, and by so doing the material and workmanship will be correct. There should always be a completely tested and assembled suction and discharge valve kept ready for immediate use. Then in the event of valve trouble the compressor can be pumped out and the new valve put in with the least loss. The defective valve is then repaired and put in stock. The foregoing gives a general idea of the care of compressor valves. To make the subject clear, the different styles of compressors must be explained as to their construction and the operation of the valves.

ARCTIC MACHINE

The distinctive features of the "Arctic" compressor are here described.

This machine is known as single-double-acting and embodies the features of both the horizontal double-acting and the

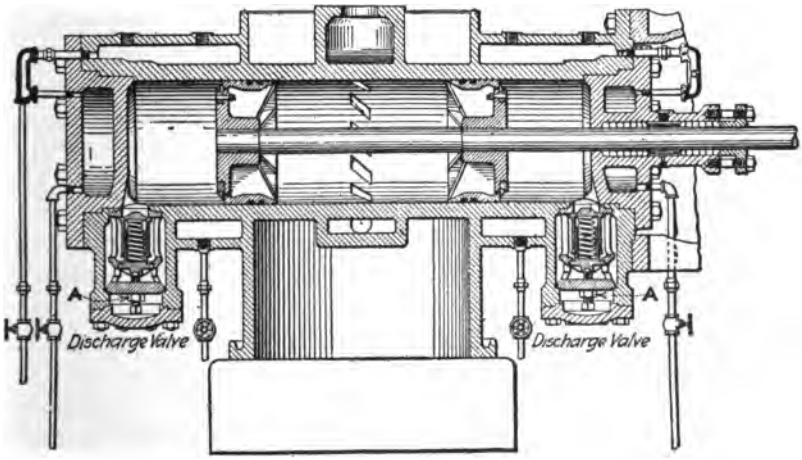


FIG. 55.—Arctic compressor cylinder.

vertical single-acting. Each end of the cylinder is practically the same as a single-acting cylinder.

There are two suction valves in the piston and one discharge valve at each end of the cylinder. The piston is in two parts, each of which forms a complete piston for one end. The

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vapor from the evaporating coils enters the cylinder between the two pistons and passes through the suction valves. Figure 55 shows the cylinder, piston and valves. Figure 56 shows the suction valves.

The suction valve and locking, or retaining, ring *A* are made of chrome-vanadium steel which has been oil- and heat-treated; its tensile strength is about 200,000 lb. The valve is light and the bearing surface so large that the wear is seldom

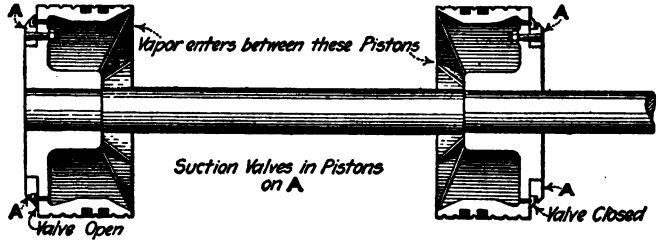


FIG. 56.—Arctic pistons with suction valves.

excessive. The valves and rings give little trouble, and any that may occur will usually be caused by some outside agency, such as scale from the inside of pipes. There are no springs, and the lift of the valve varies from $\frac{1}{8}$ to $\frac{3}{8}$ in., depending on the size of the compressor.

The retaining ring on the small machines is screwed on the face of the piston and kept tight by a taper dowel. On the large machines the ring is fastened in place by round-headed slotted capscrews sunk in to come flush with the face of the ring to reduce the clearance. Each of these capscrews is prevented from coming loose by a small screwed dowel.

A capscrew coming loose will cause a pound, as the clearance between the piston and cylinder head is not more than $\frac{1}{16}$ in. normally and often only $\frac{1}{32}$ in.

In a new plant the suction valve may cause trouble by sticking, owing to scale or other foreign matter loding between the retaining ring and the valve. It is a short and easy job to get to the valves for examination or repairs, as both the front and back cylinder heads can be removed without disturbing pipework or other connections.

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On machines up to 50 tons' capacity it takes only about ½ hr. to remove a head, examine the valve and be ready to run again. An extra retaining ring and valve should always be kept on hand for emergency use.

The discharge valves (see Fig. 55) are contained in cages and located at the lowest point at each end of the cylinder. This location and the downward opening of the valves insure the least possible danger from accident caused by liquid in the cylinder.

In this machine the valve cage is held to its seat by a steel bar and setscrew arrangement *A*; a loose cage or leakage at the ground joint is almost impossible. The valves are of the poppet type, the guide being screwed into the cage and the valve working on the plunger-like end. Cushion is provided by means of small holes which allow gas to pass in and out; the valve seats quickly and without shock. The fit of the valve on the guide must be such that binding will not occur. When wear takes place it is better to put in a new valve and guide rather than depend on bushings.

TRIUMPH COMPRESSOR

The horizontal double-acting Triumph compressor has distinctive features in valve construction. The valves, Fig. 57, are contained in cages that fit into port openings in the cylinder head. There is a ground joint between the cage and head. Instead of the valve bonnet holding the cage to its seat as in some compressors, there is a ring screwed into the head casting with two setscrews which hold the valve cage to its seat.

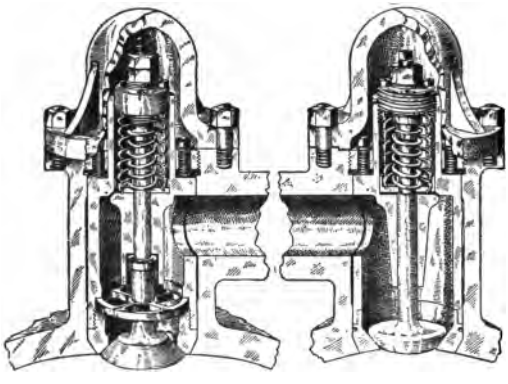
The most important feature of the valves is that provision is made for adjusting the valve-spring tension. Too great tension on the suction-valve spring will cause restricted passage and prevent the cylinder from filling with vapor at the pressure in the suction line. The higher the back pressure the greater the tension necessary on the suction-valve spring and *vice versa*.

The screwed collar that holds the valve cage prevents am-

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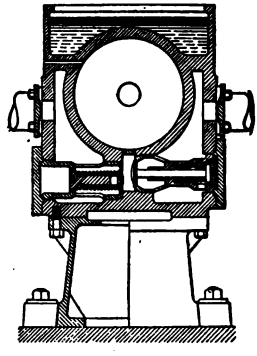
monia collecting in the valve bonnet. It also provides for easily removing the bonnet and for adjusting spring tension while the machine is operating. There is liability of ammonia leaking by the ring or valve stem, so when the bonnet is to be removed, it is advisable to slack off just a little on the nuts and allow the gas to escape.

The tension of both the working and cushion spring is adjusted separately by the collars—the inside collar for the cushion spring and the outside one for the working spring.



THE VALVES OF THE TRIUMPH COMPRESSOR

FIG. 57.—Triumph compressor valves.



END SECTION DE LA VERGNE MACHINE

FIG. 58.—De La Vergne compressor valves.

One turn of these collars is usually sufficient for a variation of about 5-lb. pressure. After adjusting the tension, the collars are secured by the screws, which act as keepers. After a little practice these valves can be made noiseless when operating under almost any pressure conditions.

Danger of the suction valve dropping into the cylinder is reduced by the safety collar, which fits in the cage and through which the valve stem passes. Above this safety collar is a collar turned on the valve stem. The stem can break above this collar with no serious results.

The discharge valve is fitted with the same arrangement for spring adjustment as the suction valve.

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DE LA VERGNE COMPRESSOR

The departure from the non-clearance machines is plainly seen in the construction of the De La Vergne compressor. On other machines the makers have done everything possible to reduce the clearance space.

As seen in Fig. 58 the suction and discharge valves are below the cylinder and there is a space between them. With



FIG. 59.—De La Vergne suction valve.

this construction there is no possibility of a broken valve getting into the cylinder. The space between the suction and discharge valves is just sufficient for the suction valve to open to its full extent. Both valves have gas cushion instead of buffer springs, so spring trouble is minimized. Details of the valve construction are shown in Figs. 59 and 60.

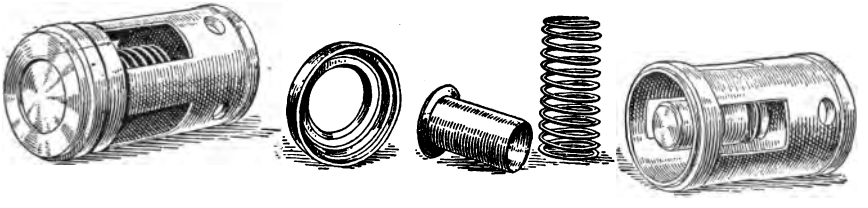


FIG. 60.—De La Vergne discharge valve.

FRICK AND YORK COMPRESSORS

The vertical single-acting machines in most general use are the Frick and the York. Figures 61 and 62 show the details of each respectively. The vapor from the evaporating coils enters the cylinder at the bottom and passes through the suction valve on the down stroke of the piston. On the up stroke the ammonia is discharged through the valve in the

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center of the safety head. On large units there are sometimes two or more discharge valves.

The suction valve, Fig. 61, is really a part of the piston and is assembled in a cage held in place by countersunk slotted screws *A*. These screws must be a tight fit at the threads and

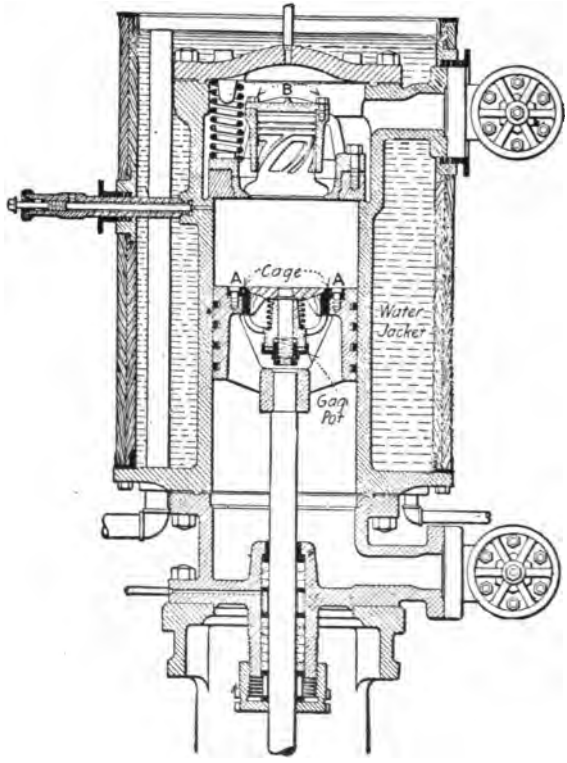


FIG. 61.—Frick compressor cylinder.

must be set up very tight to prevent their coming loose. The joint between the cage and the piston has no gasket, it being a ground joint. If, from any cause, this joint leaks and a quick repair job is necessary, a gasket of $\frac{1}{32}$-in. pure rubber or thin lead can be used, but the joint should be ground tight.

The lift of the valve, also of the closhing cusion, is adjusted by the gagpot arrangement and locknut at the bottom end of

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the valve stem. The spring should have just sufficient tension to support the weight of the valve.

The safety head on this compressor is not a discharge valve and must not be allowed to act as such. This head

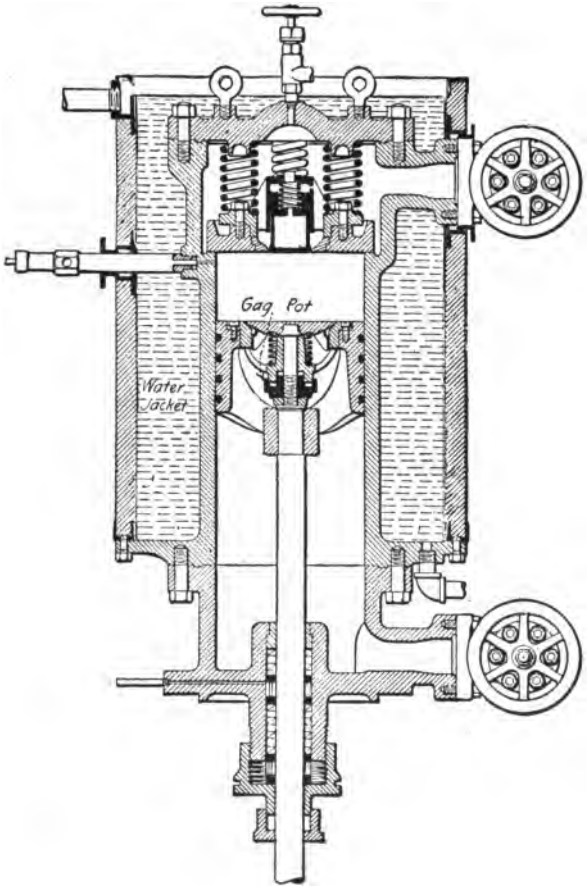


FIG. 62.—York compressor cylinder.

forms a ground joint at the top of the cylinder walls and will leak in a short time if the head moves up and down.

If leakage does occur, it will be found quicker and cheaper to insert a boring bar and face the top of the cylinder wall than

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to grind with emery and oil. A gasket can be used in emergency, but it is not advisable. To make a temporary job, a lead gasket is cut to a good fit and thick iron washers are used under the springs.

The most common trouble with this type of compressor, Fig. 61, is caused by the suction-valve cage screws coming loose. This is noticed by a slight pound when the piston passes the top center. The loosened screw will come in contact with the safety head, causing it to lift from its seat. When this slight pound is heard, do not increase the clearance to get rid of it, but lift the head and find the cause.

The discharge valve is contained in a cage fastened in the safety head by capscrews or studs and nuts *B*. A gasket can be used here, but a ground joint is best.

Leakage of the suction valve or of the joint between the suction-valve cage and the piston is first noticed by the frost melting from the suction line and the line becoming quite warm. This heating is caused by the hot gas being allowed to pass from the top to the bottom of the cylinder. A thermometer inserted in the suction line close to the compressor is the best thing to give notice of defective valves. The thermometer will show the increased temperature when there are no other indications of leakage.

Figure 62 is similar to Fig. 61 in design.

LATE DESIGN OF COMPRESSOR VALVES

The past two years have seen a wonderful advance in compressor design along high-speed lines. Due to the demand for synchronous motor-driven compressors, it has been necessary to speed up the compressors. The old design of heavy poppet valves has not been able to fulfill operating requirements. Light disc, waffer and feather valves are now largely used.

RACING AMMONIA COMPRESSORS

Racing of engine-driven ammonia compressors may be due to either engine or compressor trouble. A competent engineer should have no difficulty in locating the cause if it is in the

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engine, but when the trouble is in the compressor it is not so easily determined at times. In every case it will be found to be produced by some faulty action of the valves. If a suction valve sticks, the machine is practically unloaded for that part of the revolution, as the gas in the cylinder rushes out again through the open valve. The machine in this case will make a lunge at this part of the revolution, and for every revolution the valve sticks open. A sticking discharge valve will have about the same effect, except that the lunging or irregularity of movement will be more violent owing to the discharge pressure backing up into the cylinder.

When the valves stick only intermittently, the problem becomes still more puzzling and will, under certain conditions, produce the most violent racing. Compressor governors are rather sluggish; any large variation in the load will require some little time for readjustment of the speed. If the machine overspeeds, as it will if any of the valves stick open, it will require a few revolutions to bring the speed back to normal through the action of the governor. If, now, when the governor is at the highest point and consequently admitting the minimum quantity of steam to the engine, the sticking valve suddenly lets loose and throws the full load on again, the speed will drop far below normal before the governor finally get it under control again and brings it up to normal speed. If the valve happens to stick again at the moment the governor is at its lowest point and admitting a large amount of steam to the engine to bring it up to normal speed, the speed may become exceedingly dangerous before it is finally checked by the governor. If the valve happens to stick at the critical moment several times in succession, it may wreck the machine, unless the governor is more sensitive than the average compressor governor.

Many engineers get busy with the indicator as soon as anything like this happens. This is hardly worth while if a little judgement is used, and it takes too much time. If a discharge valve sticks, the discharge header and the valve on this end will get hotter than on the other end, because the gas that is discharged from the end that is working properly will rush into

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the end with the sticking valve, and as there is no new gas taken in, it will heat this end. In a short time the frost will also come off the suction valve on this end. If the suction valve sticks, the frost will come off the valve and the discharge will get cold on this end, as there is no gas passing out through it.

By remembering this and then using the same reasoning that one would use in locating a defective valve in a water pump, one should have little difficulty in quickly determining the cause of the trouble. For example, if the machine lunges on the forward stroke, it must be either the discharge valve on the head end or the suction valve on the crank end. The temperature of either of these and the frost on the suction valve will soon indicate which one it is. The defective valve can also generally be located by the sound of the gas rushing out through it.

However, if one is in doubt it is usually safer to use the indicator in preference to taking the valves out in a hit-or-miss search.

SUCTION AND DISCHARGE STOP VALVES

An ammonia compressor was shut down each night, and the suction valve was closed and the discharge stop valves left open for safety when starting. The compressor discharge valve and piston leaked a little, and this put head pressure on the stuffing boxes and resulted in considerable loss of ammonia, as the stuffing boxes would not hold that pressure, though they would hold the back pressure when considerably slacked off.

Another compressor was shut down at night and the only main liquid valve from the receiver closed. There was some liquid in the coils on starting, but that is better than putting head pressure on a stuffing box not built for it. Close both suction and discharge stop valves if a shutdown is going to be long enough to make it worth while closing either. But it is more important to get the discharge valve shut than to shut the suction if only one valve is closed. A rod will cool after the machine is stopped and will contract enough to leak with a high pressure on it even if the gland is tightened when first

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shut down. If the discharge valves are always shut and everyone knows it, they will be opened before starting. If not regularly shut, there is liability of someone starting with them shut.

The approved and safest manner of handling the compressor at such times is to close the suction and discharge valves after having pumped back to zero pressure. Make this a rule. Forget that you may shut down for only half an hour or less. If the suction and discharge are closed and the suction valve is tight, the machine may be started slowly and run full speed without danger, though it will do no work. When a compressor runs for a while the ammonia rods become warm and expand, and if the machine is shut down and either the suction or discharge valves are left open the stuffing boxes will leak ammonia.

Tightening on the stuffing boxes, when soft packing is used, will perhaps stop the odor. After starting up again it is natural that the rods will become hot and spoil the best packing if the stuffing box is not loosened. On the other hand, there are many ammonia compressors packed with metallic packing, which must be left alone.

In one plant there were two compressors, running only one at a time, and even then could afford to shut down for several hours at night. The custom was to leave the suction and discharge valves open. The compressor stuffing boxes were packed with soft packing. The suction and discharge valves (not stop valves) in the compressor were in first-class condition, and one was not annoyed by leaky stuffing boxes when the system had been pumped back properly.

How should the stop valves be left when the machine is shut down for the night? The best method is to shut off the liquid valve at the receiver a little before the machine is shut down and pump the expansion coils and liquid line down to zero, or at least until the machine gets thoroughly hot. Now shut the water off the condenser and stop the machines, shut the valve in the discharge line at the condenser, open the discharge check-valve bypass, close the suction stop valve and open the bypass or cross-connection from the compressor discharge to

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the suction line, letting the gas in the discharge line as far back as the condenser escape into the suction pipe, reducing the pressure to that of the suction side. Now close the discharge check-valve bypass and the discharge stop valve. Leave the valve in the bypass or cross-connection open until the machine started in the morning.

In this way there can never be more than suction pressure on the machine. There will be three valves in the discharge line between the condenser and the machine, and one of these will generally hold, preventing leakage from the condenser to the suction side through the machine. If a careless operator forgets to open the valves before starting in the morning, the open bypass valve will give sufficient relief to prevent damage and the whistling of the discharge through the restricted passage will give warning of the condition. This may seem rather elaborate to some operators who are accustomed to shutting down the machine the last thing before going home and then hurriedly leaving as soon as the machine stops; but this way takes but little more time, and there is no danger of an accident and no trouble with leakage.

The suction stop valve should always be closed as well as the discharge valve, because there is just as much danger of doing damage by leaving this valve wide open when starting as there is by leaving the discharge valve shut. Modern compressors all have relief valves for over-pressure, but if the suction valve is left wide open when the machine is started, there may be in the suction line a slug of liquid which will be swept along in a body into the compressor. This slug would put a strain on the whole machine and might blow a gasket or start a serious leak somewhere in the system.

SHOULDERS ON COMPRESSOR PISTON RODS

In the operation of ammonia compressors, especially those with a deep stuffing box, one finds that after one or two seasons' run the packing has worn a shoulder on the rod—usually two shoulders, one at the lower and another at the upper end of the rod, the distance between being the rod travel through the

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packing in the stuffing box. These shoulders will punch the top and lower rings of packing and destroy them in a short time, and practically prohibit using a metallic or semimetallic packing without having the rods turned true.

To get another season's run out of the machine without turning off the rods, have cast-rion rings made for the top and

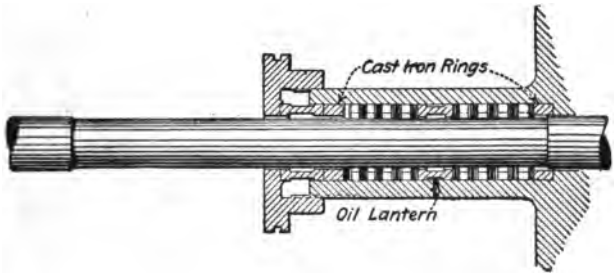


FIG. 63.—Cast iron rings in stuffing box.

bottom of box, the outside diameter of the ring fitting neatly in the bore of the stuffing box, and the inside diameter of gland $\frac{1}{32}$ in. larger than the large diameter of the rod. The rings prevent the shoulders on the rod coming in contact with the packing. Such an arrangement appears in Fig. 63.

CHANGES TO STUFFING BOX OF AMMONIA COMPRESSOR

Stuffing boxes on the older types of ammonia compressors usually were bored with beveled bottoms, and the packing faces of the glands were also beveled, the aim being to wedge the packing against the rod. This type of stuffing box on the reciprocating rod simply puts a greater tension on the outside diameter of the packing, while the rod surface of the packing is pulled either into the gland or the compressor. Machines of today are built with square-faced glands and square bottoms to the stuffing boxes.

In case a metallic or semimetallic packing is to be inserted in one of these beveled boxes, it is necessary to first put in fill rings so that square faces are produced by which the packing may be tightened. A difficulty in making these fill rings is

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that the angle a as shown in Fig. 64 may be 15 or 30 deg., or whatever the designer may have seen fit, and as it is frequently difficult if not impossible to measure this angle, a standard fill ring to meet all these varying angles is desirable.

This is obtained by making one face of the fill ring rounded or on a radius. With this type of ring this rounded face will always find a point of tangency and will pull up square.

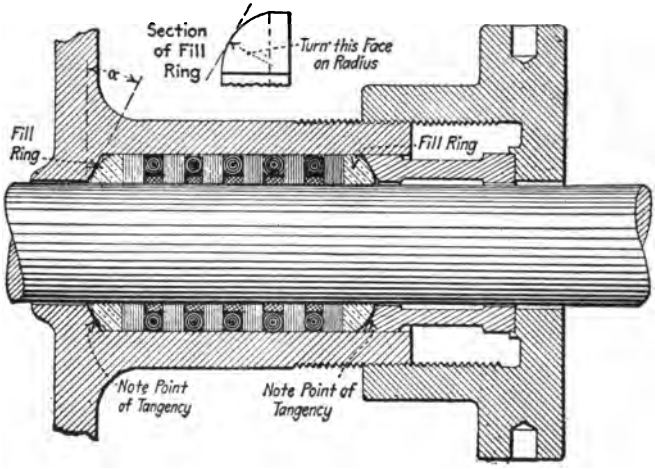


FIG. 64.—Section of stuffing box.

SELECTION OF LUBRICANT

In the selection of an oil for use in the ammonia compressor it is important that the selection be made with a view to securing one with the highest flash-point, greatest viscosity and lowest congealing point. The best is none too good as regards the points mentioned, and a question of price should not enter into the proposition at all, as the effect caused by the use of an inferior grade of ice-machine oil will offset the difference in price many times over.

The engineer should get samples of as many grades of ice-machine oils as he can from the local dealers and have analyses made of them and make his selection from the information so derived. If it is not convenient to get this information from a laboratory, he should insist on the dealer giving it to him.

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A good grade of paraffin oil, with the wax as completely removed as possible, will make a good ice-machine oil. The more paraffin wax the refinery removes from the paraffin oil, the lower will be the congealing point, which ranges somewhere around 15 to 18 deg. F. below zero.

The engineer should never under any circumstances allow engine oil to be used in the compressor or on the compressor piston rod.

As a guide to the engineer who is not familiar with the physical properties of ice-machine oils, the table below gives the essential properties of some good oils.

PHYSICAL TESTS OF OILS

Kind	Specific gravity	Viscosity	Flash, degrees Fahrenheit	Fire, degrees Fahrenheit	Congealing point, Fahrenheit
Arctic ice-machine oil.	0.8920	3.93	372	399	—19
Ice machine.....	0.8746	3.93	368	392	—24
Ice machine.....	0.8912	3.225	320	360	—20
	0.9258	4.18	330	360	—20

In the application of ice-machine oils as a lubricant, the writer has failed to find a single case in 30 years of actual operating experience where it was necessary to pump oil directly into the compressor cylinder.

Twenty years ago nearly all of the well-known ammonia-compressor builders equipped their machines with oil pumps and piped them to pump oil into the compressor cylinder direct, as well as into the stuffing box between the packing; but today oil is pumped to the stuffing box only, and in many cases a sight-feed oil cup is placed over the piston rod, allowing the oil to be fed in drops, with a moderate feed, which is found to be ample lubrication for both packing and compressor.

The oil that adheres or sticks to the rod is carried inside and is wiped off inside the compressor cylinder. Engineers have experimented along these lines considerably and have yet to report a single case in which, where the oil cup was used, the

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compressor has shown signs of wear due to insufficient lubrication.

The only excuse for an oil pump being connected to the stuffing box at all is to give the packing a "shot" of oil to stop the rod from leaking, when the packing has been allowed to remain in use so long that it will not hold, or a quick change of temperature caused by overfeeding of the evaporating coils.

The engineer will find by an examination of the system at the end of the season's run that there will be "oil aplenty" in the evaporating coils and will find also that it will be a good paying investment to clean them out thoroughly at least once a year.

EMERGENCY REPAIR OF PISTON ROD

One engineer who was somewhat new to the operating of a compressor allowed an open end wrench to wedge between the

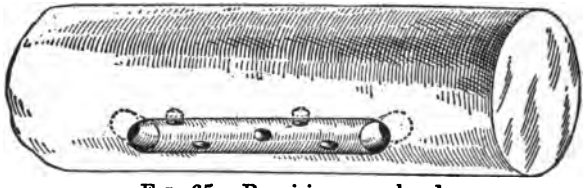


FIG. 65.—Repairing scored rod.

piston rod and stuffing-box stud. As a result a score was gouged in the rod $\frac{1}{8}$ in. deep, $\frac{3}{8}$ in. wide and 6 in. long. Not having any other way to repair the rod, he cut the groove wider, and drilled several holes along the groove. A small amount of tin was run into a mold and after cooling was peened into this groove, the drilled holes assisting in anchoring the tin. The method is shown in Fig. 65. The compressor was started and ran along without further trouble. This is essentially an emergency repair and in many cases might not stand up.

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CHAPTER IV

THE AMMONIA CONDENSER

In the direct system of refrigeration there are three parts to the cycle—the compressor, the condenser and the so-called expansion coils. Of these three the condenser is the simplest and perhaps the least understood. The condenser is only a device to abstract heat from the ammonia (much simpler than the apparatus used for steam condensation) by the use of cooling water. The compressor superheat is first removed, then the latent heat of liquefaction, and then some heat is removed to reduce the temperature of the liquid ammonia. The object of the following is to bring forward the theory of the condenser in a simple manner, to show what the condenser must do, and finally to explain why inherently some designs of condensers must be inferior to others.

Now it must be remembered that the discharge gas from the compressor has a temperature between 140 and 240 deg. F. approximately, and that the heat in this gas above the temperature of about 80 deg. F. is called compressor superheat. The gas when superheated is dry and behaves differently from saturated ammonia, which may have more or less liquid ammonia always present in the form of a mist. The action of the condensers is first to remove the superheat and then to remove the latent heat of liquefaction; and while this heat is removed, the condensing water is heated 5 or 10 deg. more in temperature, depending on the relative amount of water being used for cooling. This action can be shown diagrammatically in Fig. 66, where the upper line shows the action of cooling the ammonia from 200 to 70 deg. and the lower line shows the action of heating the water from 60 to 70 deg. F. and the arrows show the relative direction of flow of the liquids, which is counter current, or opposite in direction, in this case.

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The heat removed by the water is equal to the refrigerating duty performed plus the heat equivalent of the work of compression done by the compressor.

In another form of condenser the ammonia and the water flow in the same direction, as in the case of the old atmospheric type shown diagrammatically by Fig. 67. Here the coldest water comes in metal contact with the hottest ammonia and the condensed ammonia comes in metal contact with the hottest

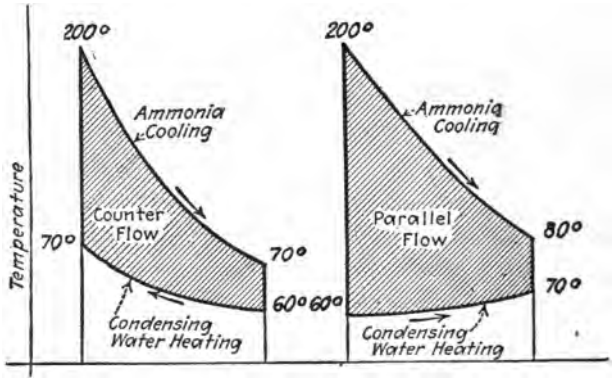


FIG. 66.—Counter flow condenser diagram.

FIG. 67.—Parallel flow condenser diagram.

water. Now it must be remembered that in heat-transfer apparatus, such as condensers, it is necessary to have a temperature difference between the two substances and that heat will not run uphill, but only from the hotter to the colder substance. In consequence it will be noticed in Fig. 67 that the liquid ammonia is cooled only to 80 deg., whereas in Fig. 66 it was 70 deg., thereby giving in each case a final temperature difference of 10 deg. between the two surfaces of the condenser.

In calculating condenser surface we use the external surface of the pipe, assuming that every square inch (top as well as bottom) is equally efficient. The metal is used only as a container for the ammonia. Heat is transmitted from the vapor or superheated ammonia to the metal, then through the metal to the water and is carried away by the water. Now suppose some air or inert gas collects on the inner walls of the

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condenser. This may easily occur, and does occur in every condenser. Then this foreign gas forms a film or skin on the metal and insulates the metal, Fig. 68A. The reader will remember that the air films make the most perfect insulator known. The effect of this is to decrease the rate at which heat will be transmitted through the condenser. By this is meant the number of heat units which can be conducted through the

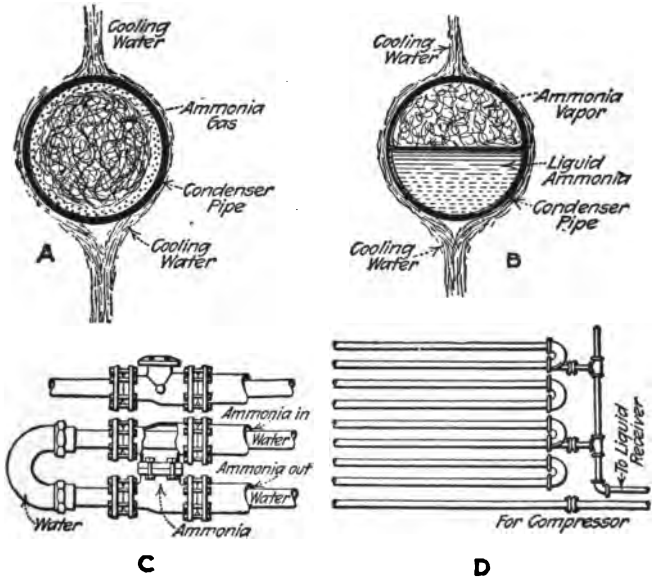


FIG. 68.—Action of condenser.

metal walls for every square foot with a temperature difference of one degree, during a time of one hour, as it is usually expressed. The value for this so-called coefficient of heat transmission may be 200, but if a gas film is present on the pipe surface it may be reduced to 50 or less.

As was mentioned, the vapor film is always present on both sides of the pipe. Its baneful action may be diminished by scouring it away by making the substances on the two sides of the pipe move vigorously. A vigorous current of ammonia vapor along the pipe has the same action as water running

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down a mountain brook—it scours the sides and removes this gas film. Therefore it is evident that if the condenser is operated so as to increase the velocity of the vapor in the condenser, the efficiency of the condenser will increase and more work can be obtained from it. However, to get this velocity of the vapor work must be done on the vapor by the compressor so that the advantage is not all clear gain.

Again, some designs of condensers allow the pipes to fill part way with liquid ammonia. This is particularly true of the old atmospheric and the so-called flooded condenser. Referring to Fig. 68*B*, it will be seen that the lower half is filled with liquid ammonia. That the accumulation of a liquid, as in this case, practically nullifies the action of that part of the condenser can be seen by anyone who has studied the action of the steam boiler. In this case the useful part of the condenser is reduced in proportion to the amount of submerged surface. Of course some heat is transmitted through the liquid, but the action is very sluggish as compared with the other portion of the pipe.

What then are the requirements of a condenser to get the best heat transfer? It must be free from air or foreign gases and it should have a velocity flow of the vapor which will tend to clean the metal surface. It must be free of oil or scale on both the inside and outside surfaces. It must be arranged so that the liquid when precipitated will be readily and quickly drained off. And finally, as economy of the refrigerating plant—as regards both head pressure and refrigerating effect of the condensed ammonia—is dependent on having as low a temperature of the liquid ammonia as is possible, the condenser should be designed on the counter-current principle.

Now, what condensers are best suited for all these requirements. If good condensing water were available, the double-pipe condenser (counter current) would answer every requirement. Foreign gases are removed by purging. The scouring action is obtained by the construction (Fig. 68*C*). The only objection is that the lower pipes are likely to be partly filled with liquid ammonia, but this is partly compensated by the fact that the liquid is cooled as near the tempera-

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ture of the incoming condensing water as is practical. The design, however, is costly, there are a number of joints to keep tight, and water tending to accumulate sediment cannot be used.

One of the oldest designs, and one that answers the requirements of an efficient condenser, is the so-called bleeder type of atmospheric condenser. This is counter current (Fig. 68*D*), the superheated gas entering at the bottom, and as it works its way upward and is condenced, it is drained off to the receiver. If properly constructed, there should be little or

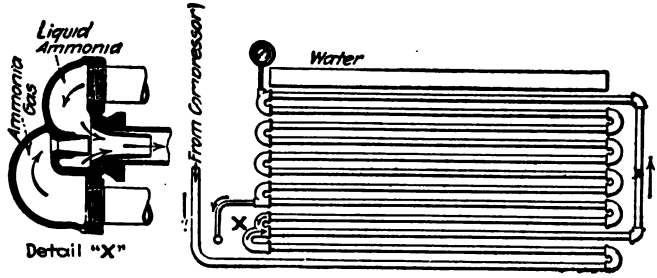


FIG. 69.—Flood condenser action.

no accumulation of liquid, and the design is such as to be very inexpensive to make. The flow is toward the upper header, where the inert gases are readily purged. As the cooling water is always on the outside, the scale or sediment may be cleaned readily. The only real objection is that the liquid ammonia is not precooled, but is drained off as it is condensed.

Finally, as regards the flooded condenser: The real object of such a condenser (which attempts to use condensed liquid ammonia to remove the compressed superheat by a kind of jet condenser action) is to try to overcome the poor heat transference from dry, superheated ammonia to the cooling water. If the ejector effect, Fig. 69, is automatically controlled to suit the load, the condenser will operate all right, although the condition shown in Fig. 68*B* will always be present. Experience has shown that when adjusted to the conditions then prevailing, the flooded condenser will work very well, but that

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any change may make the head pressure increase greatly. Tests have been published attempting to show that no advantage was derived by the flooded condenser, and of two evils most engineers prefer a low operating cost to a low first cost. Many engineers hold that there is no advantage in the flooded condenser which the bleeder type cannot equal, also that the latter is the best all-around condenser made at the present time.

CONDENSER CAPACITY

As regards the capacity of an ammonia condenser, the condition is similar to that of a steam boiler of the water-tube type. We usually rate such a boiler at 10 sq. ft. of heating surface per boiler horsepower; yet there are cases where such boilers have developed 2, 2½, 3 or even more boiler horsepower for every 10 sq. ft. of heating surface. It depends on the condition of the water, the tubes and the ability to burn fuel under the tubes. In like manner the condenser may be able to do cooling at the rate of 12, or 10, or even 6 sq. ft. per ton of refrigeration, and yet we would not be justified in using such small surfaces. The efficiency of the condenser may drop off, some stands may be laid down for repair, or perhaps storage volume may be required for the liquid ammonia during times when repair work on the low-pressure side is required.

The most important function of the condenser is the determination of the condenser pressure, which determines directly the work done by the steam engine or electric motor. If no air were present, or decomposed oil or ammonia, then the condenser or head pressure would be determined by the temperature at which liquefaction took place. And the temperature of liquefaction is determined directly by the amount of cooling showered over the condenser and its temperature. As an example, we may say that in acquiring a ton of refrigeration, the liquid ammonia in the expansion coils will absorb heat at the rate of 200 heat units (200 B.t.u.) per minute. Additional heat, amounting to about 60 heat units, is added during compression, so that the total heat to be removed by

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TABLE VII.—GALLONS CONDENSER WATER PER TON REFRIGERATION

Condenser pressure, pounds per square inch gage	Corresponding temperature, degrees Fahrenheit	60 deg. F. water				60 deg. F. water				80 deg. F. water				90 deg. F. water			
		Range, degrees F.	Water per ton of refrigeration, gallons per minute			Range, degrees F.	Water per ton of refrigeration, gallons per minute			Range, degrees F.	Water per ton of refrigeration, gallons per minute			Range, degrees F.	Water per ton of refrigeration, gallons per minute		
			Suction pressure, pounds gage				Suction pressure, pounds gage				Suction pressure, pounds gage				Suction pressure, pound gage		
			15	20	25		15	20	25		15	20	25		15	20	25
126.4	75	10	2.90	2.85	2.80												
131.4	77	12	2.40	2.35	2.30												
136.6	79	14	2.05	2.00	1.95												
141.8	81	16	1.85	1.80	1.75	6	4.87	4.80	4.70								
147.2	83	18	1.65	1.60	1.55	8	3.67	3.60	3.55								
152.7	85	20	1.47	1.45	1.41	10	2.95	2.90	2.85								
158.3	87	22	1.35	1.32	1.30	12	2.45	2.40	2.35								
164.1	89	24	1.25	1.22	1.20	14	2.10	2.05	2.00								
170.1	91	26	1.15	1.13	1.10	16	1.90	1.85	1.80	6	5.00	4.90	4.80				
176.2	93	28	1.08	1.05	1.03	18	1.68	1.63	1.60	8	3.75	3.70	3.65				
182.6	95	30	1.00	0.99	0.97	20	1.50	1.48	1.45	10	3.00	2.95	2.90				
189.1	97	32	0.95	0.93	0.90	22	1.38	1.35	1.32	12	2.52	2.48	2.43				
195.7	99	..	....	....	....	24	1.28	1.25	1.22	14	2.13	2.10	2.06				
202.5	101	..	....	....	....	26	1.17	1.15	1.13	16	1.92	1.85	1.83	6	5.10	5.00	4.90
209.5	103	..	....	....	....	28	1.10	1.07	1.05	18	1.70	1.67	1.63	8	3.83	3.75	3.70
216.5	105	..	....	....	....	30	1.02	1.00	0.98	20	1.53	1.50	1.48	10	3.07	3.00	2.95
223.7	107	..	....	....	....	32	0.95	0.93	0.92	22	1.40	1.37	1.35	12	2.57	2.51	2.47
231.1	109	..	....	....	....	..	....	....	....	24	1.30	1.28	1.25	14	2.19	2.13	2.10
238.7	111	..	....	....	....	..	....	....	....	26	1.20	1.16	1.15	16	1.95	1.90	1.88
246.5	113	..	....	....	....	..	....	....	....	28	1.12	1.08	1.06	18	1.74	1.69	1.65
254.5	115	..	....	....	....	..	....	....	....	30	1.05	1.02	1.00	20	1.56	1.53	1.50
262.7	117	..	....	....	....	..	....	....	....	32	1.00	0.98	0.95	22	1.43	1.40	1.37

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the cooling water in the condenser is about 260 heat units per ton of refrigeration. This has to be taken away by the cooling water, which is heated in consequence. This could be done by $\frac{260}{10}$ = 26 lb. (3.1 gal.) of water per minute heated 10 deg. in temperature, 19½ lb. (2.34 gal.) when heated 15 deg., or 13 lb. (1.36 gal.) when heated 20 deg. Table VII shows what can be obtained in practice.

DETERMINATION OF HEAT TO BE REMOVED

It is of course impossible to give in any table the actual amount of heat which the condenser must remove per pound of ammonia handled per minute. To obtain this in B.t.u. for any set operating conditions recourse is had to the tables showing the "Properties of Ammonia."

In compressing the ammonia vapor in the compressor it is, in these calculations, assumed that the compression is adiabatic; *i.e.*, that no heat is given up nor received except the heat due to the work of compression. In case the discharge temperature and pressure are known the heat Q to be removed in condensing the discharge vapor is $Q = h_1 + L + (t_1 - t_2)S - h_2$. Where h_1 is the heat of the liquid, L is the heat of vaporization, t_1 is the temperature corresponding to the pressure, and t_2 is the actual discharge temperature and S the specific heat of ammonia, here taken as .65 and h_2 is the temperature of the liquid ammonia in the receiver. As example, suction pressure 30-lb. absolute, suction temperature −.58 deg. F., discharge pressure 180-lb. absolute discharge temperature 120.78 deg. F. and receiver temperature 70.40 deg. F. Consulting the ammonia tables,

$$\begin{aligned} h_1 &= 64.6 \text{ B.t.u.} \\ L &= 494.1 \text{ B.t.u.} \\ t_1 &= 89.4 \\ t_2 &= 120.78 \\ S &= .65 \\ h_2 &= 121.1 \end{aligned}$$

then

$$Q = 64.6 + 494.1 + (120.78 - 89.78).65 - 42.5 = 536.59$$

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which means that under these conditions 536.59 B.t.u. must be removed by the condenser cooling water for each pound of ammonia.

CAPACITY OF AMMONIA CONDENSERS

In Table VIII below is given the amount of cooling water required per ton of refrigeration in a single-section double-pipe condenser under the conditions given. From this table values of different sizes of condensers may be figured.

TABLE VIII.—CAPACITY OF AMMONIA CONDENSERS

Capacities and horsepower per ton refrigeration of one-section counter-current double-pipe condenser, 1¼-in. and 2-in. pipe, 12 pipes high, 19 ft. outside water bends, for water velocities 100 to 400 ft. per minute. Initial temperature of condensing water, 70 deg. F.

High Pressure Constant

Condensing water				Capacity in tons refrigeration per 24 hr.	Condensing pressures pounds per square inch	Horsepower per ton refrigeration		
Velocity through 1¼ in. pipe feet per minute	Total gallons used per minute	Gallons per minute per ton refrigeration	Friction through coil, pounds per square inch			Engine driving compressor	Circulating water through condenser	Total Engine and water circulation
100	7.77	1.16	1.69	6.7	185	1.71	.0012	1.7112
150	11.65	1.165	3.05	10.	185	1.71	.002	1.712
200	15.54	1.165	5.08	13.4	185	1.71	.004	1.714
250	19.42	1.18	7.89	16.4	185	1.71	.006	1.716
300	23.31	1.24	11.41	18.8	185	1.71	.009	1.719
400	31.08	1.30	20.51	24.	185	1.71	.016	1.726

Capacity Constant

100	7.77	0.777	1.69	10.	225	2.04	.0008	2.0408
150	11.65	1.165	3.05	10.	185	1.71	.002	1.712
200	15.54	1.554	5.08	10.	165	1.54	.005	1.545
250	19.42	1.942	7.89	10.	155	1.46	.009	1.469
300	23.31	2.331	11.41	10.	148	1.40	.016	1.416
400	31.08	3.108	20.51	10.	140	1.33	.038	1.368

NOTES.—Above tables are based on the heat transmission obtained for various velocities of water, as averaged up from York Manufacturing Company's tests on double-pipe condensers.

The horsepower per ton is for single-acting compressor at 15.67 lb. suction pressure.

The friction in water pump and connections should be added to water horsepower and to total horsepower.

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GETTING CONDENSER COILS TO WORK EQUALLY

At first temperature readings should be taken once every two hours; that is, the temperature of the outgoing water from each separate coil should be taken. If the water from one coil shows by its comparative coldness that too much water is being circulated for the work accomplished, the valve on that coil is closed a little. But do not expect to notice a difference during a few minutes or even one half an hour. Do not change the position of the valves oftener than once every two hours.

The best way to handle this temperature taking and water regulating is to have a card or suitable piece of paper tacked up close to the regulating valve of each coil. On this card mark down the temperature every two hours and also make a note as to what change is made in the position of the valve. In this way the valves will be handled intelligently and guesswork done away with.

After the proper flow of water through each coil has been found and the valves set, it will not be necessary to make further changes. The effect of this even temperature of the overflow water will be noticeable in the uniform liquid level in the liquid receiver. Also the regulation of the expansion valves will be easy as the temperature of the liquid will not vary.

In a distilled-water ice plant the separate outlet from each coil can be piped into a wooden or metal trough and the water for the steam condensers can be taken from this. The trough must be provided with an overflow so as to take care of any water that is not needed for the condensing of the steam. When a cooling tower is used, the outlets from the condenser coils are extended to the top of the tower.

Now that a method of water regulation is in use, the next step toward good results is free exchange of the heat from the ammonia to the water, *i.e.*, the condenser must be kept clean of mud and scale.

OPENING AMMONIA CONDENSERS

Removing the caps from the inclosed tubes of a double-coil ammonia condenser may be easily and safely accomplished

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with a home-made device such as that illustrated in Fig. 70. A plate ½ in. thick is cut out as shown to about 4½ by 6 in., with slots to receive the ends of the U-bolt, which is placed under the cap and the nuts tightened on the plate to give a pull on the cap. The nature of the operation is clearly shown in the illustration. A given number of caps can be removed in about one-fourth the time required by the use of a hammer and chisel, and the gaskets are seldom spoiled.

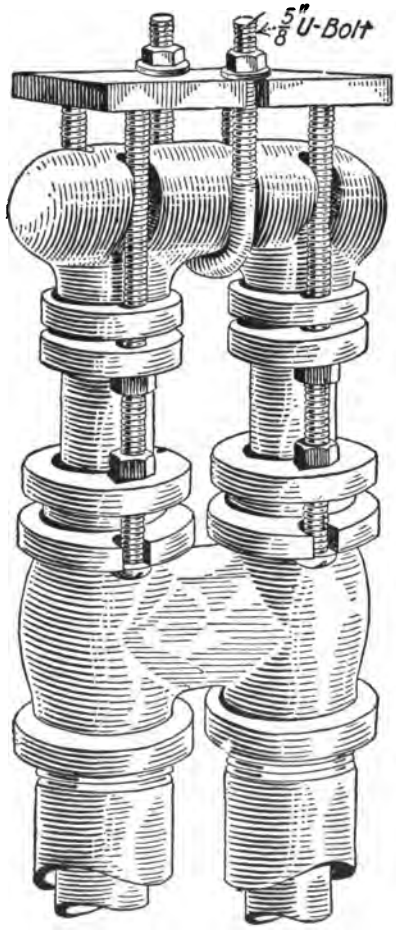


FIG. 70.—A clamp to open condenser joints.

CONDENSER AND BACK PRESSURE

The lower the condenser pressure and temperature and the higher the back pressure and temperature, the more economical will the plant give service.

The proper pressures for the condenser and the low-pressure side of the system is a question which every engineer has to solve for his particular plant. In each plant the conditions will be a little different, and the pressures which will give the highest economy will depend on the temperature of the cooling water, the amount of water supplied, the square feet of cooling surface in the condenser per ton capacity, the temperature it is desired to maintain in

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the refrigerator and the amount of non-condensible gases present. If the surface in the condenser is known to be correct and there is little trouble from non-condensible gases, then the following data may be taken as an approximate guide. Figure 71 gives the proper condenser pressures for different temperatures and quantities of cooling water.

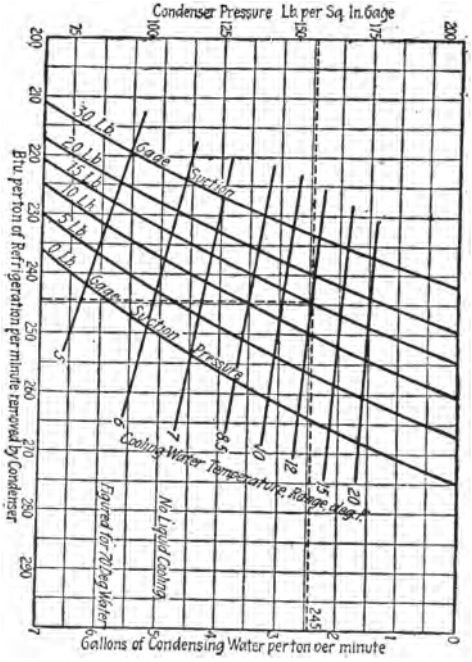


FIG. 71.—Chart for determining quantity of cooling water required for various conditions of pressures and temperatures of water and ammonia.

After determining the proper pressures to be carried in any plant it is important that these pressures be maintained. The easiest and probably the best way to do this is to install thermometers in the suction and discharge pipes of the compressor. Each temperature corresponds to a certain pressure, and by marking or indicating in some way the proper height of the mercury column it is easy to see at a glance whether temperatures are above or below normal. If one can afford

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it, it is well also to use gages in connection with the thermometers to check the latter.

In the spring and fall, when the temperatures of the air and cooling water are lower, it is possible to reduce the range in pressure from that used in the hot summer months. To obtain the highest economy, adjustment of pressure should be made at least three times during the season. It is a simple matter and requires only ordinary intelligence on the part of the operating engineer. A reported case in point is where the output of an ice plant was increased over 50 per cent by raising the temperature of the compressor discharge from 104 to 240 deg. F. The latter proved to be the proper temperature for the existing conditions.

THE PURGE TANK

Every engineer knows that from time to time it is necessary to purge the condenser to remove the inert gases from the system. Those inert gases may be air or decomposed ammonia, and their effect is to increase the head pressure. For instance, if the condenser pressure is 180 lb. and the condensed ammonia is at a temperature of 70 deg. F. (corresponding to a pressure of 114.5-lb. gage) then the difference, 65½ lb., is the extra amount against which the compressor has to work and which is a needless waste of energy.

The usual manner of purging is simply to open a valve at the top of the condenser and allow the gases to pass into a pail of water. Careful engineers make a point of stopping the compressor and keeping the cooling water on the condenser from half an hour to several hours before purging, the idea being that all the ammonia will be condensed and only the inert gases will remain. This is an improvement on the first method, yet it is becoming better known every day that a large amount of ammonia is lost through the purge valve.

This loss of ammonia is due to the fact that the law of partial pressures and the law of diffusion hold for ammonia as for a mixture of perfect gases. This means that the only advantage of running water on the condenser and stopping

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the compressor is to cool the mixture of gases and condense out *some* of the ammonia. We all know how it is that in cooling atmospheric air we first reach the dew point (where the air is 100 per cent humid) and then any further cooling will cause precipitation of moisture. One way of *drying* air is to cool it, as is done in the low-pressure can agitation system using stationary air pipes. So, in the condenser, by cooling the mixture of the gases some ammonia is condensed and the remainder is weaker in ammonia content. But at no time will there be a stratum formed with air and inert gases on top and ammonia gas on the bottom. The mixture will be about the same in all parts of the condenser. Therefore, when any purging is done, ammonia as well as air is purged. But can any improvement be made?

In the small-capacity plant it is not worth while to do much of anything more, but the large plants should try to prevent this loss. It is not possible, by means of the condensing water, to cool sufficiently the gases being purged, but they can be refrigerated by means of a cooling coil in an extra header at the top of the condensers into which all of the stands are connected. In purging off, the procedure would be, as before, to stop the compressor and continue the cooling water over the condenser for some time, then open the expansion valve on the freezing coils in the purge header. The gas should be cooled to about 32 deg. F. or lower. The gain to be expected may be seen by the following example, with a few assumptions.

Assume a condenser that has 1,000 cu. ft. of gas volume, cooling water is at 70 deg. F., and half of the condenser volume is purged. (These conditions are rather extreme, however.) The condenser pressure is taken as 180 lb. gage. As the pressure in a mixture of gases is proportioned to the volumes present, we may say that the gas in the drum is composed of $\frac{104\frac{1}{2} + 14.7}{180 + 14.7} = \frac{129.2}{194.7} = .66$ per cent of ammonia and $100 - 66 = 24$ per cent of inerts. Suppose we cooled the gas in the drum to 32 deg. F. The condenser pressure will remain practically the same, but the percentage of the mixture will change, as at 32 deg. F. the vapor pressure of ammonia is

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61.9-lb. absolute. Therefore, $\frac{61.9}{194.7} = 31.8$ per cent = relative volume of ammonia present, and $100 - 31.8 = 68.2$ per cent = relative volume of inerts.

In the first case 66 per cent of 500 cu. ft. = 330 cu. ft., or $\frac{330}{2.3} = 143$ lb. of ammonia purged; in the second case 31.8 per cent of 500 = 159 cu. ft., or $\frac{159}{4.82} = 33$ lb. of ammonia purged. The difference, $143 - 33$ = approximately 110 lb. ammonia saved. At 30 cts. per pound the gross saving is $33.00.

COOLING WATER

In general there are three ways of cooling water where reuse of cooling water is required—cooling ponds, spray ponds and cooling towers. Each of these methods has its advantages in special cases.

The cooling pond can be used in locations where a natural body of water is available adjoining the plant. The water is cooled by being allowed to be brought in surface contact with the air, and the evaporation (and cooling of the water) depends on the relative temperature of the air and on the humidity. The tendency would be for the pond to assume the temperature of the air, both due to giving up heat to the air and also due to the difference of the *saturation pressure* of the water and saturation pressure of the water vapor in the air. Roughly, the amount of cooling in *still air* for usual conditions may be taken at from 60 to 80 B.t.u. per square foot of surface per hour, requiring about 225 sq. ft. per ton of refrigeration.

COOLING TOWERS

The action of the cooling tower depends mainly on the giving up of heat by the hot water to the air. The tower is usually a high structure, circular in section, with its interior filled with some sort of baffle work designed to break up the water into fine sheets or sprays. The hot water is delivered to the top of the tower while a strong current of air is injected into the

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tower at the base by means of a forced-draft fan. The air and water come into intimate contact within the tower, and the water is cooled and the air heated. The cooled water from the tower is collected in a basin at the foot, and the condenser water is taken from this supply.

SPRAY PONDS

The action of the spray pond depends on the cooling of water by evaporating part of it and thus taking from what was left the heat of vaporization necessary to do the evaporating. Conduction and convection were not relied upon for cooling. This effect is present, also, to a small extent in any cooling tower as some of the water evaporates and carries away heat in the form of steam.

Cooling in the modern spray pond is accomplished, as nearly everyone knows, by spraying the warm water through nozzles designed to break the water up into very fine particles or spray, thus aiding the evaporation tendency. A great deal of experimental work has been done by American engineers in the perfection of these nozzles. At the present time there are several different designs on the market, all of which apparently give good results.

Sprays are arranged in rows and placed a certain distance apart, depending upon the system employed. Theoretically, the farther we place the sprays apart the better the cooling. In practice we are, of course, limited in this respect to the land available and the cost of the pond. These spray heads are attached to the main pipe line either in clusters of five or six, or alone. The rows of sprays are generally placed closer together than the nozzles in a single row. This arrangement is supposed to create a draft between the rows and thus produce better cooling. The general direction of the current of air is upward, however, and this consideration would seem to be somewhat theoretical.

EFFICIENCY OF SPRAY PONDS

About the only way to express the cooling of water in terms of a per cent efficiency is to compare the amount of cooling

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that has resulted under a given set of conditions with that which would have resulted had perfect cooling conditions prevailed. By perfect conditions we mean the cooling of the water down to the wet-bulb temperature, which is, of course, the lowest possible to obtain. We may express this in terms of an equation as follows:

$$E = \frac{T_1 - T_2}{T_1 - Tb} \times 100$$

where

E = per cent efficiency;
T_1 = initial temperature of the hot water;
T_2 = final temperature of the water after cooling;
Tb = temperature of the wet bulb.

So many factors affect this value that the reader had better consult the paper by Prof. C. C. Thomas on this subject, read before the American Society of Mechanical Engineers at the December, 1917, meeting. The value of Tb may be taken from any humidity table for the conditions under consideration. For rough calculations the reader may use a value of $E = 70$ per cent and not be far off.

Many people do not realize that the maximum cooling obtainable in a spray pond is dependent on the relative humidity. This factor is more important than the air temperature because it is possible to cool water below the air temperature on days of low humidity. The Spray Engineering Co. quotes tests that show a cooling of 24 deg. F. below the air temperature. The relative humidity stood at 19 during these tests, which is, of course, unusually low. It is more common practice to be able to cool the water to within a few degrees of the wet-bulb temperature. If the engineer will consult the weather reports for the locality in which the pond is to be located and get from them an average value of the humidity for the last few years, he will be able to draw an intelligent conclusion upon the amount of cooling he can obtain.

If the difference between the temperature of the water to be sprayed and the wet-bulb temperature at the given humidity is more than about 20 deg. F., it will generally be necessary to

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spray the water twice in order to get it down to the desired point. This is undoubtedly due in large measure to the impracticability of placing the spray heads very high above the surface of the pond. There is also a limit to the fineness of the spray which it is possible to obtain, and it is now generally agreed that fineness of spray is more important than a long passage of the water through the air.

WHEN DOUBLE SPRAYING IS NECESSARY

When it becomes necessary to spray the hot water twice in order to obtain the desired low temperature, two possible cases arise. One of the most common methods is to build two ponds adjoining, with a single wall or bank between them. The water is taken from the hotwell by the pumps and delivered to the sprays in one of the ponds. It is then collected in a sump, usually in one corner of this first pond, and piped back to the pumphouse, where another pump delivers it to the sprays of the second pond, where the final cooling is accomplished. Here again the cooled water is collected in a sump and piped to the condenser pumps, which completes the cycle. Such a system must be arranged so that either pump can deliver to either pond and so that either pond can deliver to the condenser pumps. Either pump must be so arranged that it can draw water from the condenser sump. It will usually be advisable to arrange one spare pump in such an installation, so piped and valved that it can take the load of either of the two others when they break down or are in need of overhauling. In winter it will usually be unnecessary to use more than one pond. The climatic conditions locally will greatly affect the details of the design.

MIXING HOT AND COOLED WATER BEFORE SPRAYING

Another method, when double spraying is necessary, is to mix the warm water from the condensers with some of the cooled water from the pond before it is sprayed at all. This lowers the temperature of the water in proportion to the amount of cooled water added. This mixed water is then carried to the

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pumps and forced by them to the sprays and requires but one spraying to reach the same low temperature obtained by the use of two ponds. Let us further analyze this arrangement and note its advantages. In the first place, but one pond is required, which saves the dividing wall. But one pipe line need be laid from the pumphouse to the pond. If this distance is long, the saving will be considerable. We have succeeded in eliminating the heavy moisture which was due to the spraying of the warm water. In cold weather part of the sprays may be shut off if properly valved. The reader must not draw the conclusion that a smaller pond is possible with this arrangement. Generally speaking, this would not be true, because the water that is taken from the pond to cool the warm water in the mixing well is really sprayed twice, and so the pumps and nozzles have the same work to do as if the double spraying were done in two separate ponds. If the warm and cooled water were mixed in equal proportions, this statement would be exactly true.

It may be found, however, that it is unnecessary to add equal parts of the cooled water in order to reach the desired temperature. Valves should be so arranged that the operator can adjust this mixture to meet the daily conditions, and he should at no time add more of the cooled water than is necessary to get results from the pond. A thermometer showing the temperature of the condenser feed water, together with a chart showing the temperature possible under the various conditions of temperature and humidity, is all that is needed. Perfect control is then always possible, and even temperature of condenser water means even and maximum efficiency the year around. It may happen that more than an equal quantity of cooled water will have to be added to get results. This would appear on hot, humid days, in summer. If the spray system is flexible and has been designed for a little excess capacity, this will be possible and your plant will be able to maintain its vacuum as well as on any other day. This condition would really mean that part of the water was sprayed a third time.

It is of interest and may be of some value to the reader to

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express in the form of an equation the relation existing between the temperature of the hot, cooled and mixed waters and the quantities involved in each case. We will use the following notation and assume that the specific heat of temperatures employed. This assumption is almost exactly true.

Let

Q = quantity of hot water from condensers in gallons per minute;

Q' = quantity of cooled water from pond in gallons per minute;

Q'' = quantity of mixed water going to spray pumps in gallons per minute;

T = temperature of hot water in degrees Fahrenheit;

T' = temperature of cooled water from pond in degrees Fahrenheit;

T''' = temperature of mixed water passing to spray pumps in degrees Fahrenheit.

The following relations must hold:

$$Q'' = Q + Q' \tag{1}$$

and $8.36\ Q''T''' = 8.36\ Q'T' + 8.36\ QT$ or $Q''T''' = Q'T' + QT$. Therefore

$$T''' = \frac{Q'T' + QT}{Q''} = \frac{Q'T' + QT}{Q' + Q} \tag{2}$$

The quantity 8.36 used in these equations is the number of B.t.u. required to raise one gallon of water one degree. Note, however, that this value 8.36 cancels out in all members of the equation. Also note that the quantity has been considered in gallons per minute, but that the equations are entirely independent of the unit of quantity, as well as of the unit of time, as long as the same unit is used throughout. In other words, the quantity may be in cubic feet or pounds, and per minute, per second per hour. It makes no difference in the final result, which is the obtaining of the temperature of the mixed water.

It may well happen that the designer or operator desires to know the value of Q'; that is, the quantity of cooled water from the pond which will be necessary to obtain a certain

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temperature of the mixed water. Equation (2) may be rewritten for this purpose in the following manner:

$$Q' = \frac{Q(T - T'')}{(T'' - T')} \qquad (3)$$

Such an equation would prove of value when the line from the pond to the hotwell was equipped with a meter. It would aid the operator in quickly arriving at the desired amount of cooled water necessary to obtain results.

DEPTH OF POND AS AFFECTING EFFICIENCY

There is another factor which affects the efficiency of a spray pond, and that is the depth of the water employed. Of course, the more shallow the pond the greater will be the area exposed to radiation and conduction in proportion to the body of water. This would seem to be an asset. Experience has shown, however, that a deep pond is the more desirable, when possible, because of the lag in temperature between day and night made possible by the smaller surface exposed to the heat of the sun and the larger body of water. The pond will cool down considerably during the night and will not rise to as high a value during the day in the deep pond. This lag will keep the average temperature of the pond during the twenty-four hours several degrees below the average for the shallow pond. This is a decided gain. On the other hand, the deep pond costs more to build and the designer must strike a balance between the interest on his investment and the gain to be realized by the lower temperature of the water. This may not always be an easy thing to do. Practice has shown that a pond should be between 6 and 8 ft. in depth to give good results.

Spray ponds may be made completely of concrete, although a cheaper method is to use as little concrete as possible and make the walls and bottom of puddled clay. The pond should be made to extend mostly below the surface, rising not more than 1 or 2 ft. above. The exact location is usually determined by the head desired on the pumps. The sump, or collecting basin where the water is taken back, should be

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made of concrete and screened to prevent foreign matter from passing to the pumps and condensers. The concrete around the sump must be well bonded into the clay to prevent leakage around or under it. A spray pond placed well above the ground is a menace to surrounding property, because in case of failure a flood with its accompanying damages results. Provision should be made whereby the pond may be completely drained and cleaned. An overflow must also be provided sufficient in size to take care of the maximum amount of water which could ever reach the pond. In arriving at the size of the overflow, it must be remembered that the water is coming to the pond under some head and flowing from it through the overflow under practically no head, and that under these conditions a larger-sized pipe must be used for the overflow than is used to bring the hot water in.

PUMPING SYSTEMS FOR SPRAY-POND SERVICE

Let us look now for a moment into the pumping-system side of the spray pond. The centrifugal pump is now almost universally used in this service. The reasons for this are its cheapness, good efficiency, flexibility and sureness of operation. It is simple in construction and therefore causes little trouble. Foreign matter will, to some extent at least, pass through it without injury to the pump, and there are no valves to get out of order.

There are some fundamental points to be observed in a centrifugal-pump installation. But one pump should ever be run on one suction. If two or more are made to use the same intake pipe, trouble will be experienced if it is desired to start one with the other running. The one that is running will tend to lose its suction. The same rule applies, although to a lesser degree, with the discharge pipe. Check valves may be employed to overcome this, but when at all possible each pump should have a separate discharge line. If the pump is placed above the level of the supply pond, provision must be made for priming the pump with a separate pipe from some other source. Frequently, the city water supply may be

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utilized for this purpose. When such conditions exist a foot valve must be placed in the suction line near the pump, otherwise the priming water would all run back to the well. It is usually good practice to place a gate valve in the suction line when the water comes to the pump under head. The pump is then accessible for repairs.

THE PUMP SHOULD NOT BE PLACED TOO HIGH

It may be laid down as a good working principle to place the pumps low enough so that they may at least be filled by gravity when not running. This makes priming automatic, and with the gate valve closed on the discharge and the suction valve open, the pump is always primed and ready to start. Check valves should always be placed in the discharge line whenever the head worked against is large or when more than one pump discharges into one main. The check valve must be placed next to the pump and the gate valve beyond for two reasons: First, to prevent water-hammer from injuring the pump; and, second, to make the repair of the check possible without stopping the whole system. The suction pipe which extends into the water in the well must be well inserted to prevent a whirlpool action from drawing air into the suction line. If very much air enters in this way, the pump will lose its water and will have to be stopped and reprimed. This applies, of course, only in the case of pumps that are placed above the water level.

In concluding, the relative merits of spray ponds over the cooling towers might be enumerated:

1. They produce better and more uniform cooling.
2. They are cheaper in first cost and much cheaper to maintain.
3. Much lower head is required on the pumps, which represents a saving in power.
4. There is no air to handle or fans to get out of order.
5. Better control and more flexible is afforded.
6. They give a better supply of water for fire purposes.

The tower has one advantage of being available for use in close quarters.

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CHAPTER V

THE EVAPORATING SYSTEM

EXPANSION COIL SYSTEMS

Storage Systems.—Formerly when most of the refrigerating systems were engaged in the manufacturing of ice only, the tank coil system was used almost exclusively. In this system coils of pipes are placed in a brine tank in which are suspended the cans of water to be frozen. The ammonia liquid flows from the receiver through the expansion on regulating the valve into these coils. The boiling or evaporating of the liquid ammonia produces the refrigerating effect. Many engineers are under the impression that the ammonia passes through the expansion valve as a gas. To the contrary, but little refrigeration would be secured if this were true; the liquid enters the coil as a liquid.

Of late years the flooded system has come into general use. With this system usually a large accumulator is used. The coils are filled with liquid, the vapor passing off of one end into the accumulator which, being half filled with liquid, maintains a constant head on the coils.

In cold-storage plants several systems are employed; the direct-expansion and indirect-brine systems being more generally used. As the name indicates, with the direct-expansion system the ammonia is fed into the storage room coils.

With this system, where ammonia is allowed to boil in the cold-storage rooms there is danger at all times. The pipe lines may become corroded, or split due to imperfect welding, or fittings may be broken by accident. The result is that some or all of the ammonia in that part of the system may be emptied into the room. There have been accidents to liquid receivers where the anhydrous ammonia ran in streams into the street.

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Besides the danger to life and commodities, there is also the matter of the initial charge of ammonia. In a large system of long supply and return pipes, or extensive refrigerating piping, the amount of the initial charge has to be very heavy, which requires constant care and repair to maintain the piping tight at all times. The result is that, under certain particular conditions, the so-called brine system is best for the refrigerating piping.

THE BRINE SYSTEM OF INDIRECT REFRIGERATION

In the brine system of indirect refrigeration, the high-pressure side is the same as in the direct-expansion system. The low-pressure side consists of a brine cooler, usually of the shell and tube type similar in construction to a steam condenser. The brine is a non-freezing solution of sodium or calcium chloride of such density as will not freeze at the temperature carried in the cooling system. On account of the lower freezing temperature of calcium brine, it will be found in most service where zero deg. F. brine is required.

The brine system, then, is really an additional unit in the system, where the brine is kept cool by boiling ammonia and the cold-storage rooms or other refrigerating applications are kept cold by the brine. The ammonia system is self-contained, and the ammonia piping may be kept constrained to a single room. The brine, however, must be kept circulating constantly by means of a centrifugal or reciprocating pump, and all exposed piping must be well insulated. The brine line need only be full-weight steel pipe with ordinary fittings, valves and cocks. The system is laid out in many ways like an ordinary hot-water heating system.

ADVANTAGES OF THE BRINE SYSTEM

The great advantage of the brine system, other than those already noted, is in the ability to store up energy. With the direct-expansion system the compressor must be operating constantly during the live-load period. If this is not done, the pipes will soon lose their frost and the room temperature, or

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the commodities being cooled, will begin to increase in temperature. With the brine system, this is not true, as "cold" may be accumulated in the brine and this may be circulated while the compressor is shut down. In fact, the example may be carried considerably farther in certain kinds of service that are intermittent, as ice-cream making, milk and drinking-water cooling, the chill room of the packing house, or where refrigeration is required suddenly.

In cases of periodic load the brine-storage tank becomes an advantage. In such a case refrigeration may be stored by cooling a large volume of brine through as large a temperature range as is practical. The compressor may be operated 24 hr. of the day, thereby utilizing a much smaller machine than would be required if no brine was used and the compressor capacity had to be capable of the maximum rate of cooling. It will be seen that by means of brine the peak load is distributed during the whole day of 24 hr., whereas the load may last only 3 hr. Energy is accumulated in the same sense as in the case of an engine flywheel.

DISADVANTAGES OF THE BRINE SYSTEM

There are, however, certain disadvantages in the use of brine. The first cost and the operating cost are usually greater. The brine pump has to be operated by independent power. In addition, and more serious than anything else, are the losses incident to the use of brine. It will be remembered that the brine is first cooled in a brine cooler, and the ammonia may be required to boil at 16 deg. F. to carry brine at an average temperature of 26 deg. F. Likewise, the air to be cooled in the cold-storage room could be kept at a corresponding temperature of 36 deg. F., thus allowing 10 deg. difference of temperature in each case. The capacity of the compressor would be determined by the temperature of the boiling ammonia, which would have been 26 deg. F. in the case of direct expansion. In the brine system, therefore, it is necessary to boil the ammonia at considerably lower pressure than with the direct expansion, thereby costing more

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per ton of refrigeration for the power input, and reducing the capacity of the compressor.

The brine required for a given condition may be calculated easily. The "heat capacity" of a certain volume of brine depends on the product of the density (specific gravity) and its specific heat (heat units required to raise 1 lb. 1 deg. F.). For instance, the cooling effect of 100 gal. of brine in raising 4 deg. F. would be

$$\underset{\text{Degrees range}}{4} \times \underset{\text{Number, gallons}}{100} \times \underset{\text{Weight, gallons}}{8\tfrac{1}{3}} \times \underset{\text{Specific gravity brine}}{1.2} \times \underset{\text{Specific heat brine}}{0.7} = 2{,}700 \text{ B.t.u.}$$

assuming 1.2 for the specific gravity and 0.7 for the specific heat of the brine. These values change with each concentration of brine solution.

If it is required to find the amount of brine per minute necessary to provide one ton of refrigeration with a five-degree range of temperature, the calculation becomes

$$200 = \text{No. gallons} \times 8\tfrac{1}{3} \times 1.2 \times 0.7 \times 4$$

$$\text{No. gallons} = \frac{200}{8\frac{1}{3} \times 1.2 \times 0.7 \times 4} = \frac{200}{28} = 7.14 \text{ gal. per min.}$$

REGULATION OF PRESSURES AND TEMPERATURES

Those responsible for the operation of refrigerating machines are aware that to reduce the temperature of a cooler or freezer to zero or below, it is necessary to lower the suction pressure to the compressor; but as many engineers do not understand why this is necessary, an explanation will be welcome, as this book is for the benefit of the operating engineer, and not the college graduate or professor, although practical knowledge of operation is beneficial to the college man as well.

The table of the properties of saturated ammonia will show the engineer the boiling point of liquid ammonia for any required suction pressure, and when it is required to reduce the temperature of a freezer to, say, 20 deg. F. below zero, it will be easy to understand that the temperature of the boiling liquid on the inside of the pipes that are placed in

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this freezer will have to be reduced below the temperature of the room. By referring to the tables previously mentioned it will be found that the boiling point of liquid ammonia at 0-lb. pressure; or atmospheric pressure on the suction or low-pressure gage, will be 28½ deg. F. below zero, which is only 8 deg. F. below room temperature.

From the foregoing the operating engineer will readily see why it is necessary to reduce the suction or low pressure to produce low temperatures, and also he will find a good reason for ammonia getting lost, sluggish or hiding.

When the temperature of a cooler or freezer is only about eight or ten degrees above the boiling temperature of the liquid ammonia on the inside of the expansion or evaporating coils, the engineer can expect nothing except a slow or "sluggish" evaporation. Do not swear at the man who sold you the ammonia for selling you an inferior quality or a kind of ammonia that will not mix with what you already had, etc.; but look for the cause. If the temperature is not holding or coming down as it should, the cause is generally due to slow evaporation, as already explained, a shortage of ammonia in the system, not enough pipe surface in the cooler for the product to be handled, and many other causes which we will treat later.

The evaporating coils are the boiler of the refrigerating plant, and the engineer or temperature man should watch the return end of the coil the same as he watches the water glass on the boiler.

DUTIES OF THE TEMPERATURE MAN

In large plants where there are thousands of feet of direct-expansion pipe and hundreds of expansion valves, it is necessary to employ a temperature man, whose duty it is to regulate these valves, take temperatures, etc., and report them properly made out on a temperature or log sheet, which enables the engineer to properly direct and operate the plant.

In plants where several temperature men are employed, there are times when it is necessary to "break in" new men on

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this work, and at such times there is always more or less danger of overfeeding the expansion coils and the accompanying danger to the compressor, together with alternate "freezing up" and "burning up" of the compressor, which also carries with it an unusual loss of ammonia due to expansion and contraction of the piston rod, packing, joints and connections.

Most steam plants use a separator between the engine and boiler to protect the steam cylinder from water, and for the same protection to the ammonia compressor, there should be a separator between the evaporating coils and the compressor, as there is no difference in the results other than a much greater need for a separator on the suction line leading to the compressor as the danger to life and property are many times greater in case a compressor head is knocked out, due to suffocation from ammonia gas, fire, etc., which subjects will be treated later.

It is necessary only to secure an ordinary steam separator, preferably one having a storage capacity, for liquid, so that in case of liquid being carried over to the compressor from overfeeding the evaporating or expansion coils, the storage capacity in the separator will give time for the liquid to drain back to the evaporating coils, or other means so provided for the purpose.

INSTALLING THE SEPARATOR

The separator should be installed at a point high enough above the evaporating coils to insure a speedy drainage of the separator, also the drain line should be provided with a valve to close in case of pumping out.

The drain line from the separator can also be connected to a tank located below the separator for the liquid to drain into. This tank is provided with a water jacket, with steam connection to keep the water hot for the purpose of reevaporating the liquid ammonia, the gas piped back to the suction side of the compressor; and in this way the tank serves the double purpose of converting the liquid from the separator into a gas and purifying the ammonia by separating out the oil or water that may have been taken into the system by leaks or inferior

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ammonia and the oil that works into the system from the compressor.

When the liquid from the separator is carried back to the evaporating coils, there is practically no loss from this source, as the liquid is reevaporated and will do useful work in the transfer of heat, but the beneficial results of leading the separator drain to a purifier will more than offset the gain by the other method of reevaporating the liquid in the coils.

Another system for removing liquid from the separator is to connect the drain line to an automatic drip tank, and pump which is operated with a float and returns the liquid to the expansion or evaporating coils, the operation being the same as that used in a steam plant for returning condensate to the steam boiler.

The frost on the suction line leading to the compressor does not mean that there is liquid ammonia being carried over to the machine, for the reason that the gas is in a saturated condition and is at the same temperature as that of the boiling liquid. If the boiling liquid is below the freezing point of water, it will freeze the moisture that is carried in the air which comes in contact with the cold surface of the pipe, forming a coating of frost of a thickness depending upon the saturated condition of the air, temperature of the ammonia, gas, and time.

On all horizontal double-acting compressors it is good practice to "freeze back," to the machine, as the heat of compression is much greater than in the single-acting compressor.

FEW PLANTS ARE FREE FROM DECOMPOSED AMMONIA

Ammonia (NH_3) is decomposed by heat, and it is claimed by some authorities that a complete separation of the nitrogen and oxygen takes place at a temperature of about 900 deg. F., while others claim that decomposition takes place very slowly, at a much lower temperature, ranging from 150 to 300 deg. F. There are few plants that are entirely free from decomposed ammonia. There may not be enough to noticeably affect the capacity of the plant, but nevertheless it is there in greater

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or lesser quantities, and for this reason alone, the frost should be carried close enough to cool the gases, leaving the machine to a temperature of 100 to 150 deg. F., which can easily be determined by inserting a thermometer in the discharge line of the compressor.

Another reason for holding the temperature down on the discharge line, is to prevent vaporizing of oil which, if carried to the oil trap at a temperature of from 300 to 500 deg. F., will be in a more or less vaporous or finely atomized condition and will float with the hot gas to the ammonia condenser where it will be condensed to a liquid, then go on to the evaporating coils where the oil congeals more or less, depending on the freezing point of the oil. When the coils accumulate enough of this oil it will cut down the refrigerating effect of the ammonia, as it is a good insulator, and will also stop up the feed or expansion valves to such an extent that regularity in the feeding of the coils will be impossible; often the coils stop up altogether, which necessitates a shutdown for cleaning out the system.

In large plants it is often necessary to run suction lines for a distance of 300 to 600 ft. and even longer, and in such cases it is difficult to freeze back to the compressor with any degree of regularity owing to a large number of crooks and turns, risers, drops, radiation, etc. In such cases it is more effective and less dangerous to cool the compressor by connecting a ¼-in. line from the liquid receiver and expand direct into the suction line at a location somewhere close to the compressor.

In traveling these long distances, there are considerable losses due to radiation, as the insulation is never perfect enough to prevent these losses. To successfully cool the compressor or "freeze back" through these long lines requires considerable practice. If there is oil in the system, as mentioned, it is impossible to cool the gases leaving the compressor with any degree of regularity. It will be a feast or a famine. The compressor will either get the ammonia in "chunks" or will be "burning up," together with all the other interesting conditions caused by irregularities in the handling of expansion valves. It is a great deal more safe, sane and economical to

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bring bad conditions close up under the watchful eye of the operating engineer, than to trust to Providence and the temperature man not to blow the place up.

SETTING AMMONIA EXPANSION VALVES

Setting the expansion valve on an ammonia refrigerating plant that has only a single valve is a very simple job. Generally every refrigerating engineer has some settled idea or notion about when the machine is running just right, and all he has to do when there is only one expansion valve is to keep on adjusting it until the machine works right.

Generally there is considerable difference of ideas about when a machine is working right. Some insist on carrying the frost line back so far that the cylinder of the machine gets frosted and the water freezes in the water jacket at times. Others only carry the frost line to within a few feet of the suction header of the machine.

Still others gage their ideas of the work the machine is doing by feeling of the discharge line of the machine, handling the expansion valves so that this is so hot that the hand can be barely held on it for a short time. Others gage their setting of the expansion valves and the operating of the machine by the suction pressure. Some judge the operation of the machine by the temperature of the piston rod, as determined by feeling of it at regular intervals by the bare hand.

All of these methods of judging the operation of the machine have their advantages under certain circumstances, and all of them will be taken into consideration by the operator who knows his business. The whole secret of efficient ice-machine operation can be stated in a few words: Keep the suction pressure as high as possible to do the work in the lowest temperature services and return the suction gas to the machine in a saturated, but not wet condition; or else a few degrees superheated. If the gas comes to the machine a few degrees superheated there is no trouble with liquid entering the machine and reducing the capacity as well as increasing the power requirements per ton of refrigeration.

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USE OF THERMOMETERS

The latest and scientific method of judging the operation of an ice machine is to use thermometers in both the suction and discharge lines and in connection with the suction gage adjust the operation of the machine according to the indications of these. For every suction pressure there is a corresponding temperature, if the gas returns to the machine in a saturated condition. If the gas returns more or less superheated the temperature will of course be higher than the temperature corresponding to the suction pressure.

Gages as a general rule never remain accurate for long periods and thermometers have been known to get out of order. Thermometers, however, do not get out of order near as often as a pressure gage and as a consequence it can be generally assumed that the gage is to blame if the two do not correspond.

To check the two, open up on the expansion valves until the discharge line gets cool enough so that the hand can be comfortably held on it, when under the average operating conditions there will be a small amount of liquid carried along with the suction gas into the machine and the returning suction gas will be fully saturated. Note the temperature of the suction gas and then from an ammonia table find the pressure that corresponds to this temperature. If the gage registers this pressure it is probably correct to within a fraction of a pound. If there is a difference it will be the same over the general operating range of the suction pressure, and after the difference is once noted it can be added to or subtracted from the gage reading to get the correct reading for all pressures over the entire range, which will generally be not more than 5 or 10 lb.

If the system is not provided with thermometers in the discharge and suction lines of the machines one will have to judge the operation of the machines by the feeling of the different parts by hand, the suction pressure, and the frost line on the machine. As a general rule engineers prefer to carry the frost line back to the suction valves on the machine, keeping these frosted, while the discharge remains hot. If the discharge line gets cold when the frost line is carried this far

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back let it drop back far enough, by pinching back on the expansion valves, so that the discharge line is hot enough to boil drops of water that are thrown on it, or a little cooler than this, which is about the same thing as saying that the temperature of the discharge should be somewhere around 200 deg. for average conditions. If the suction pressure is high and the condenser pressure low the discharge line will run much cooler than this under the most efficient operating conditions, and if the suction pressure is low and the discharge pressure high the discharge line may be hot enough to boil water real lively if a few drops are thrown against it. In some cases it may run up to over 300 deg.

Under no circumstances should the machine be run cold enough so that the jacket water leaves the jacket cooler than it enters. Everyone has seen many plants run in this way, year in and year out, but the owner paid for it in increased coal or power bills. As a general average rule for the beginner to follow it is advisable to carry the frost line back to the suction valves, or nearly so, keep the discharge line warm and keep the suction pressure as high as possible, and still leave enough temperature difference between the ammonia and the room in the lowest temperature rooms to keep the temperature at the desired point. What this pressure will be will depend upon the lowest temperature carried on any room on the system, and the amount and cleanliness of the pipes. Even under the most favorable conditions there should be about 10 deg. temperature difference between the ammonia in the coils and the temperature in the rooms, or the lowest brine temperature carried on the system in case brine is used. Sometimes where a room is heavily piped or a brine cooler or tank has abundant pipe it is possible to operate with a temperature difference considerably lower than this but these cases are few and far between.

COILS WITH DIFFERENT TEMPERATURES

Now as has been said before, it is a simple matter to adjust the expansion valve when there is only one valve on the system. If there are two there is twice the trouble and the amount of

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trouble that can be experienced and the number of complications that can arise increase in direct proportion to the number of expansion valves to be handled.

To illustrate this point let us assume that there are two valves to be adjusted on the system. Let us further assume that it is a direct-expansion system and that one room is to be held at 15 deg. while the other should be held at about 32 deg. To give a temperature difference of 10 deg. between the room temperature in the lowest temperature cooler and the ammonia in the coils the ammonia temperature must be 5 deg. This corresponds to a suction pressure of about 20 lb. gage and this will be the suction pressure if the system is properly operated. If the low temperature room is insufficiently piped the suction pressure will have to be carried lower than this in order to keep the room temperature down. The low-temperature room, however, will determine what the proper suction pressure will be.

In many plants however, the room temperatures are often run much higher than desired for the simple reason that they do not adjust the expansion valves so that the whole coil is working, or if there are several coils in the room, so that all the coils are working at maximum capacity.

Many operators have the habit of opening all of the expansion valves on the system about the same amount. Very often there are different sized coils in the same room, or in a number of rooms of about the same temperature, and the operator adjust each valve to about the same amount of opening. The result is that some one coil will get more liquid than it can evaporate while others will not get anywhere near enough and as a consequence only one or two coils will be working to capacity and the rest will not be doing anywhere near like what they ought to do.

Figure 72 illustrates the construction of the average expansion valve. In many respects this resembles the average boiler gage cock, and the action of the ammonia as it passes through is identically the same as that of the water that passes out from a gage cock when it is opened. As the ammonia is expanded from the higher to the lower pressure some of the ammonia is

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evaporated to cool the liquid down to the lower temperature and the mixture as it escapes from the expansion valve into the coil is a combination of liquid and gas which shoots into the coil at high velocity.

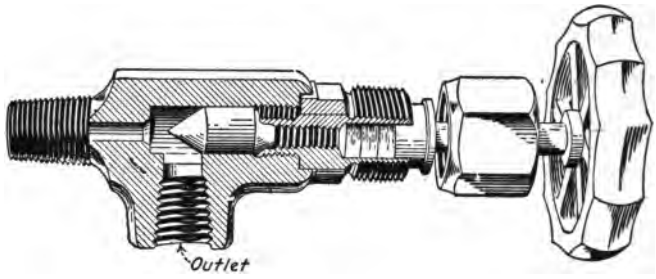


FIG. 72.—Expansion valve.

ACTION OF AMMONIA IN THE COILS

In addition the ammonia already in the coil is boiling the entire length of the coil and the liquid in the coil surges backwards and forwards somewhat in the manner of boiling water; sometimes a sudden change of temperature takes place around a coil, or the air is suddenly agitated by the opening of a door or some one working nearby and an extra surge will take place in the coil and probably throw some of the liquid into the suction line. As a consequence it will be seen that it is something of a problem to so adjust the expansion valves so that every coil is getting all the liquid that it will evaporate and still not throw any into the suction line.

One must always remember that for the same room temperatures a short coil will not evaporate as much liquid as a long one. A coil double the length of another in the same room will evaporate about twice the amount of liquid that the short one will. Also a coil badly frosted will not evaporate as such liquid as one that is fairly clean. A coil of a certain length in a room where the temperature is high will evaporate much more liquid than a coil in a room where the temperature is low. For instance, if a suction pressure of 15-lb. gage is carried the ammonia temperature will be about zero. If one of the rooms is held at 15 deg. and another at 30 deg., there

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will be 15 deg. temperature difference between the room and the ammonia in the low-temperature room and 30 deg. difference in the hightemperature room. The same length of coil in the high temperature room will evaporate about-twice as much ammonia as the one in the low temperature room and all of these things must be taken into consideration when adjusting the expansion valves.

ADJUSTMENT OF EXPANSION VALVES

For the most efficient operation one will have to size up the entire system, and then adjust the expansion valves according to the coil lengths, the room temperatures and the condition of the coils. After an approximate setting has thus been determined one will have to go along and experiment with each valve in turn until a point is reached where the frost line, discharge temperature, or temperatures as indicated by the thermometers, in case the machine is equipped with these, approach what the operator considers the proper condition for his particular plant. This will require considerable patience and experiment on the part of the operator, but he will find after a while that for a given suction pressure and room temperature there is a certain setting for each expansion valve, and once this is determined they will need only an occasional shaking up to keep them working properly. Of course, when the temperatures change or the suction pressure changes some more adjustment must be made.

Referring again to Fig. 72 it will be seen that the opening through the valve is quite restricted. In usual operation the valve is generally not opened much over one-quarter to one-half turn. As a consequence, the slightest amount of scale in the pipes, or dirt or other foreign matter mixed with oil will shut a valve off in a hurry. Pieces of packing also float around in the system and help to make trouble. As a consequence one must make regular inspection of the entire system, if all the coils are to be kept working at the proper capacity all of the time.

To get an idea of what an expansion valve sounds like when it is working properly, place your ear against a small

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water valve under high pressure and catch the sort of grinding sound it makes when pretty well throttled down. An expansion valve working properly will make practically the same sound. On a short coil, however, in a cold room it can only be opened enough so that only a gentle hissing sound is heard. Once you get used to the various sounds made according to the adjustment required for each coil you can quite easily tell when a coil is working properly or not. If then you find that the suction pressure begins to drop or the machine starts to get hot, you can generally detect the trouble-making by going up to each valve and placing your ear to it. The dead one is the trouble-maker. You can also tell by the frost line on the valve. At the point where the pressure changes from the high to the low pressure there will be a sharp frost line across the valve. If the valve is plugged the entire valve will frost over and in this case it is easy to locate the valve that is making the trouble.

One thing one must watch out for when taking charge of a new plant is that all of the expansion valves are of the regulation pattern. If the common stop valve is used, as is found in many cases, the same amount of turn on the valve wheel of this will feed a much greater amount of ammonia than the regular expansion valve. The regular expansion valve has only a small hole and needle-point valve, as will be seen from Fig. 72 and the usual stop valve has an opening several times as large as this. Furthermore there is a difference between expansion valves of different makers, so that if there are a number of makes on the job one will have to experiment with each make until their proper adjustment is mastered.



FIG. 73.—Expansion cock.

Figure 73 illustrates an expansion cock, which is often used in place of the regular expansion valve. They have the

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advantage of quick action in case they become clogged and one wants to blow them clear, and there is not quite as much as a pocket for the foreign matter to accumulate as in the valve. For close adjustment, however, many engineers prefer the valve, although for large coils where a large amount of liquid is to be fed at all times they usually prefer the cock. One advantage of the cock is that if you carry a handle around with you there is no possibility of anyone meddling with the valve, which is quite an advantage in a lot of places. Once the cock is set you take the handle away with you and unless someone improvises a handle the setting will not be disturbed until you come back.

DIRECTION OF FLOW IN BRINE COOLERS

In the shell-type cooler better results are had by allowing the brine to flow up instead of down through the cooler as is the general custom. If the cooler is so operated that superheated gas may leave it, it is better to bring the warm brine into the top, but when saturated and even wet vapor leaves the cooler, better results may be obtained by bringing the warm brine into the bottom.

Brine coolers operated in parallel: When brine coolers are operated in parallel with two brine pumps, it would be advisable to so cross-connect the pumps and coolers that either pump can be operated on either cooler independently. It is difficult to so arrange the piping connection the coolers and the pumps that the flow of brine is equally distributed between the two coolers, even though they may be the same capacity and identical in construction.

CALCIUM CHLORIDE

Use rubbers while working with it, as in breaking it up. To open a drum lay it down, beat its sides a little, stand on end, remove the top and cut through the drum from top to bottom. Calcium brine is far less corrosive than salt brine and is a better allround refrigerating medium.

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Put the broken calcium (in pieces no heavier than about 10 lb.) into the brine tank with the required water and start the brine pump, circulating the brine in the tank. If a small quantity is to be made it may be most convenient to stir the water and calcium with a piece of pipe. Make the solution a little stronger than required for the service, then dilute with water. A table of brine solution appears on pages 127 and a table of properties of calcium chloride is given below.

TABLE IX.—CHLORIDE OF CALCIUM SOLUTION

Specific gravity at 64 deg. F.	Degree Beaumé at 64 deg. F.	Degree salometer at 64 deg. F.	Per cent of $CaCl_2$	Freezing point in degrees Fahrenheit	Ammonia gauge pressure pounds per square inch
1.007	1	4	0.943	+31.20	46
1.014	2	8	1.886	+30.40	45
1.021	3	12	2.829	+29.60	44
1.028	4	16	3.772	+28.80	43
1.035	5	20	4.715	+28.00	42
1.043	6	24	5.658	+26.89	41
1.050	7	28	6.601	+25.78	40
1.058	8	32	7.544	+24.67	38
1.065	9	36	8.487	+23.56	37
1.073	10	40	9.430	+22.09	35.5
1.081	11	44	10.373	+20.62	34
1.089	12	48	11.316	+19.14	32.5
1.097	13	52	12.259	+17.67	30.5
1.105	14	56	13.202	+15.75	29
1.114	15	60	14.145	+13.82	27
1.112	16	64	15.088	+11.89	25
1.131	17	68	16.031	+ 9.96	23.5
1.140	18	72	16.974	+ 7.68	21.5
1.149	19	76	17.917	+ 5.40	20
1.158	20	80	18.860	+ 3.12	18
1.167	21	84	19.803	− 0.84	15
1.176	22	88	20.746	− 4.44	12.5
1.186	23	92	21.689	− 8.03	10.5
1.196	24	96	22.632	−11.63	8
1.205	25	100	23.575	−15.23	6
1.215	26	104	24.518	−19.56	4
1.225	27	108	25.461	−24.43	1.5
1.236	28	112	26.404	−29.29	1-in. vacuum
1.246	29	116	27.347	−35.30	5-in. vacuum
1.257	30	120	28.290	−41.32	8.5-in. vacuum
1.268	31	...	29.233	−47.66	12-in. vacuum
1.279	32	...	30.176	−54.00	15-in. vacuum
1.290	33	...	31.119	−44.32	10-in. vacuum
1.302	34	...	32.062	−34.66	4-in. vacuum
1.313	35	...	33.	−25.00	1.5 lb.

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TABLE IX.—BRINE SOLUTION
(Chloride of sodium—Common salt)

Percentage of salt by weight	Degree on Salometer at 60 degrees Fahrenheit	Specific gravity at 60 degrees Fahrenheit	Specific heat	Weight of 1 gal.	Pounds of salt in 1 gal.	Pounds of water in 1 gal.	Weight of 1 cu. ft.	Pounds of salt in 1 cu. ft.	Pounds of water in 1 cu. ft.	Freezing point degrees Fahrenheit
0	0	1.	1.	8.35	0.	8.35	62.4	0.	62.4	32.
1	4	1.007	0.992	8.4	0.084	8.316	62.8	0.628	62.172	31.8
5	20	1.037	0.96	8.65	0.432	8.218	64.7	3.237	61.465	25.4
10	40	1.073	0.892	8.95	0.895	8.055	66.95	6.695	60.253	18.6
15	60	1.115	0.855	9.3	1.395	7.905	69.57	10.435	59.134	12.2
20	80	1.150	0.829	9.6	1.92	7.68	71.76	14.352	57.408	6.86
25	100	1.191	0.783	9.94	2.485	7.455	74.26	18.565	55.695	1.00

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PUMPING AMMONIA FROM COILS

To clean the evaporating coils it will be necessary to transfer the ammonia from the evaporating coils to the condenser, and in case the coils are submerged in a brine tank the engineer should make certain that there are no leaks before pumping them out.

The brine can be removed before pumping out, but in this case air would be taken into the system. The proper thing to do is to assure oneself that there are no leaks and then proceed to pump the ammonia out of the coils before the brine is removed, as by so doing the brine will evaporate the liquid ammonia much faster. After the coils have been pumped down to as low a vacuum as possible, shut the main liquid and suction valves, and then the tank coils can be opened without any loss of ammonia due to unevaporated ammonia lying in the coils.

A sure method of cleaning these coils, after first removing the headers, is to take each one separately and connect up a steam hose of sufficient size to get a high velocity of steam traveling through the length of the coil. The heat from the steam will warm the oil, and the velocity of the steam will carry it through and discharge oil and dirt. When the oil is all out, disconnect the steam hose and attach an air hose, which will absorb and carry out the moisture remaining in the pipe, as the walls of the pipe will remain heated a sufficient length of time to evaporate the moisture and allow it to be carried out, which will leave the inside surface clean and dry. After all rubber joints have been examined, tightened and replaced where necessary, the coils are ready to be connected and put in service. In pumping out direct expansion coils, especially where the coils are in coolers where the ammonia fumes are absorbed by the stored products, extreme care should be taken to see that there are no ammonia leaks when the coils are opened.

It is well to take the extra precaution to open a line leading to the coil somewhere outside the cooler, as there may be a leaky valve, and in this case it may be necessary to put in a "blind" outside the cooler.

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To simply pump a high vacuum on the coils is no assurance that the coils are empty in the case of a direct-expansion system. Take an ammonia test tube and fill it with liquid anhydrous ammonia to the amount of 50 or 100 c.c. and allow it to evaporate in a room where the temperature ranges from 90 to 100 deg. F. Note how long it will take to evaporate. This will give the engineer an idea of how long he will have to hold the vacuum on a direct-expansion coil in a cold room for the ammonia to evaporate. The vacuum gage may show a good vacuum for an hour or two without perceptible loss after being pumped down and still leak when the coil is opened and warm air comes in direct contact with the remaining liquid inside the pipes. The better way is to make sure of hunting up the traps or low places in the pipe lines and then apply the blow torch.

TWO INDIRECT METHODS OF FINDING THE WEIGHT OF AMMONIA IN CIRCULATION IN COMPRESSION REFRIGERATION

Many different methods have been used to determine the weight of ammonia in circulation in compression refrigerating systems. Among the direct mechanical methods are the use of meters of various types and the placing of tanks, mounted on scales, between the condenser and the receiver. It is evident that these entail the installation of special and expensive apparatus and add to the already large number of valves and joints that must be kept tight.

It is proposed here to outline two non-mechanical methods of computing the weight, which require little in the way of apparatus beyond the plant proper. They will be considered separately, the shorter and simpler, which might be termed the pressure-volume method, being discussed first. The necessary apparatus additional to the plant includes thermometers to read discharge temperature at the compressor cylinders, a revolution counter (usually installed with the plant) and an indicator with which to take indicator diagrams from the compressor cylinders. The charts of the properties

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of ammonia, which accompany this discussion, complete the necessary equipment.

A brief explanation of the charts is advisable at this point. The various curves of Fig. 74 relate to liquid ammonia and to wet and saturated ammonia vapor, but do not enter the

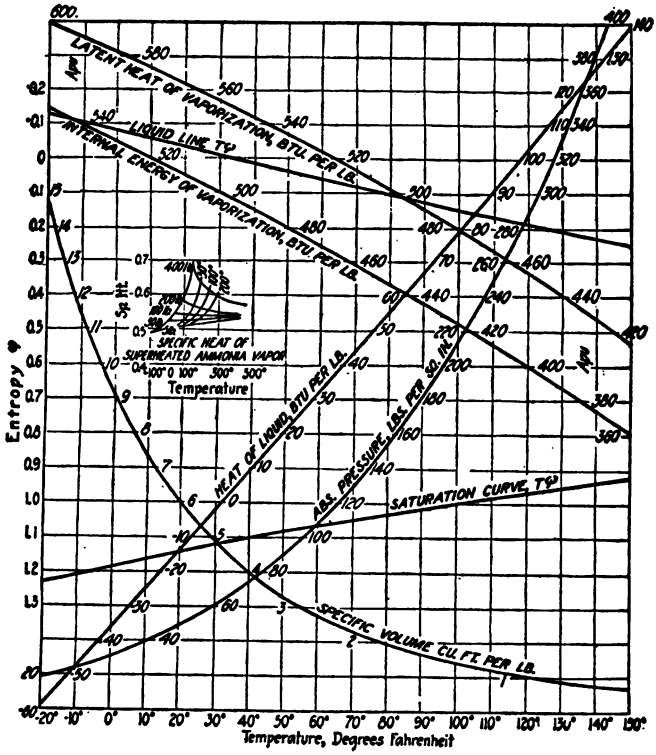


FIG. 74.—Thermodynamic properties of ammonia.

superheat field. On this chart the only curve that applies to the present problem is that of absolute pressures vs. temperatures. This curve is, of course, a means of determining the boiling points of liquid ammonia at various absolute pressures or *vice versa*.

Figure 75 deals with superheated ammonia vapor and is fundamentally a temperature-entropy chart in which degrees

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of superheat have been substituted for temperature. Superimposed on the temperature-entropy coordinates are two additional sets of lines, one of constant absolute pressures and one of constant pressure-volume (PV) products (pressure in pounds per square inch $\times$ volume in cubic feet per pound). The advantage of this arrangement will become apparent as the discussion proceeds.

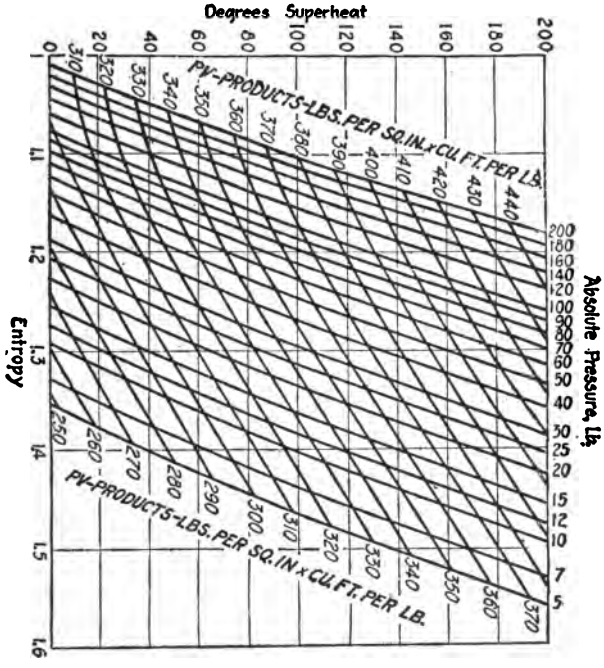


FIG. 75.—Properties of superheated ammonia vapor.

The demonstration and calculations will be made from an actual indicator diagram and actual data taken from a test run on the 15-ton York plant at Cornell University. To obtain maximum accuracy, the data should be averaged from readings and diagrams taken at intervals throughout a run of considerable length. For simplicity of demonstration one card only is used in this discussion, although the remainder of the data has been averaged from a series of readings.

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Figure 76 is an exact copy of a diagram taken from one cylinder of the two-cylinder compressor. The additional necessary data are as follows:

Barometric pressure in mercury, 29.95 in., equals	14.7 lb. per square inch
Discharge pressure, gage	68.0 lb. per square inch
Discharge pressure, absolute	82.7 lb. per square inch
Discharge temperature, degree Fahrenheit	138.5
Dimensions of compressor cylinder	8 by 12 in.
Piston displacement per revolution	0.443 cu. ft.
Scale of indicator spring	60 lb.
Revolutions per minute of compressor	52.8

The first step is to find the PV production of 1 lb. of ammonia at discharge conditions; that is, on the high-pressure side of the system. From Fig. 74 the boiling point corresponding to an absolute pressure of 82.7 lb. per square inch is 45.5

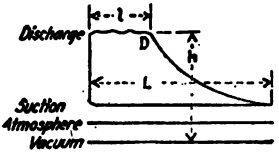


FIG. 76.—Diagram from 15-ton compressor.

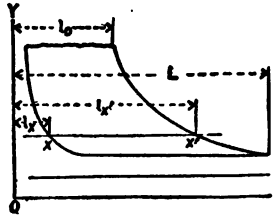


FIG. 77.—Diagram showing re-expansion.

deg. Thus the superheat at discharge is 138.5 − 45.5 = 93 deg. Turning now to Fig. 75 and following the horizontal line of constant superheat of 93 deg. to its intersection with the constant pressure line of 82.7 lb., a PV product value of 362 is determined.

The remainder of the procedure is based on measurements of the diagram taken at point D (Fig. 76), which is the point where discharge pressure is reached and ammonia begins to leave the cylinder.

The actual volume at D is $\frac{l}{L} \times 0.443$ cu. ft., in this case $\frac{0.77}{2.29} \times 0.443 = 0.149$ cu. ft.

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The diagram, Fig. 76, shows clearly that there is no reexpansion in the cylinders; that is, there is no clearance volume. Should the diagram show reexpansion, as in Fig. 81 (exaggerated), the clearance volume must be determined and allowed for. The clearance line OY (Fig. 77) may be calculated from direct measurements of the cylinder or may be graphically determined by various methods described in books dealing with steam-engine-indicator diagrams. To determine the allowance to be made, draw any line xx' parallel to the vacuum line and intersecting both the reexpansion line and the compression line. The total volume in the cylinder is $\frac{l_x}{L} \times$ (piston displacement + clearance volume), but of this the proportion $\frac{l_x}{l_{x'}}$ is due to clearance and will not be discharged at the end of the stroke. This ratio may safely be assumed to hold for the entire reexpansion line, so the true volume to be considered at point D is $\left(\frac{l_D}{L} \times \frac{l_x}{l_{x'}}\right) \times$ (piston displacement + clearance volume).

The absolute pressure at D, Fig. 76, is $h \times 60$, or $1.39 \times 60 = 83.4$ lb. per square inch. The actual PV product is therefore $0.149 \times 83.4 = 12.4$.

The weight of ammonia in the cylinder is thus $\frac{PV \text{ actual}}{PV \text{ for one lb.}}$, or $\frac{12.4}{362} = 0.0342$ lb. The weight per hour follows by multiplying by the r.p.m., or revolutions per minute, and by 60; that is, $0.0342 \times 52.8 \times 60 = 109$ lb. per hour approximately.

A similar process is applied to the other cylinder, and the sum of the two results gives the total weight in circulation per hour. In this case the total was 236 lb.

This method may be applied only when the vapor is superheated at discharge, but as this is almost universally the case in present practice, its application will not be greatly limited on that account.

It will be noticed that this procedure is based on the assumption that any change that takes place in the vapor as it passes through the discharge valve has but a negligible effect on the

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value of the PV product. This passage through the valve is evidently a wire-drawing effect and takes place at constant heat. A critical analysis of the process on this basis, using values from the tables previously mentioned, shows that this wire drawing has no perceptible effect on the PV product.

As for other sources of error, analysis shows that discrepancies in pressure due to inaccuracy of the gage or in reading will induce an error in the PV value of 10 per cent of the relative error of the pressure determination. In other words, an error of 1 per cent in pressure will cause an error of $\frac{1}{10}$ of 1 per cent in the PV value.

CALCULATING PIPE

The following lengths of 2-in. pipe are recommended by F. L. Fairbanks for rooms to be maintained at the temperatures mentioned:

1 ft. pipe to 10 cu. ft. between 10 and 32 deg.
1 ft. pipe to 40 cu. ft. between 32 and 50 deg.
1 ft. pipe to 60 cu. ft. at or about 50 deg.

Practical Rules for Direct-expansion Piping.—These values are liberal. One-ton refrigerating capacity is a liberal allowance for every 4,500 cu. ft. of cold-storage capacity held at 32 deg. F. For every ton of refrigeration allow 260 to 300 ft. of $1\frac{1}{4}$-in. pipe. Luhr suggests 600 linear ft. of 2-in. pipe per ton refrigeration for direct-expansion systems, and 750 ft. for brine systems. For storage rooms he allows 1 ft. of 2-in. pipe for 25 cu. ft. of space; for freezing rooms, 1 ft. of 2-in pipe per 7 cu. ft. space; for fish freezing rooms, 1 ft. of 2-in. pipe per 2 cu. ft. of space. In Fig. 78 are given three charts showing the amount of pipe required for various expansion systems and room temperatures.

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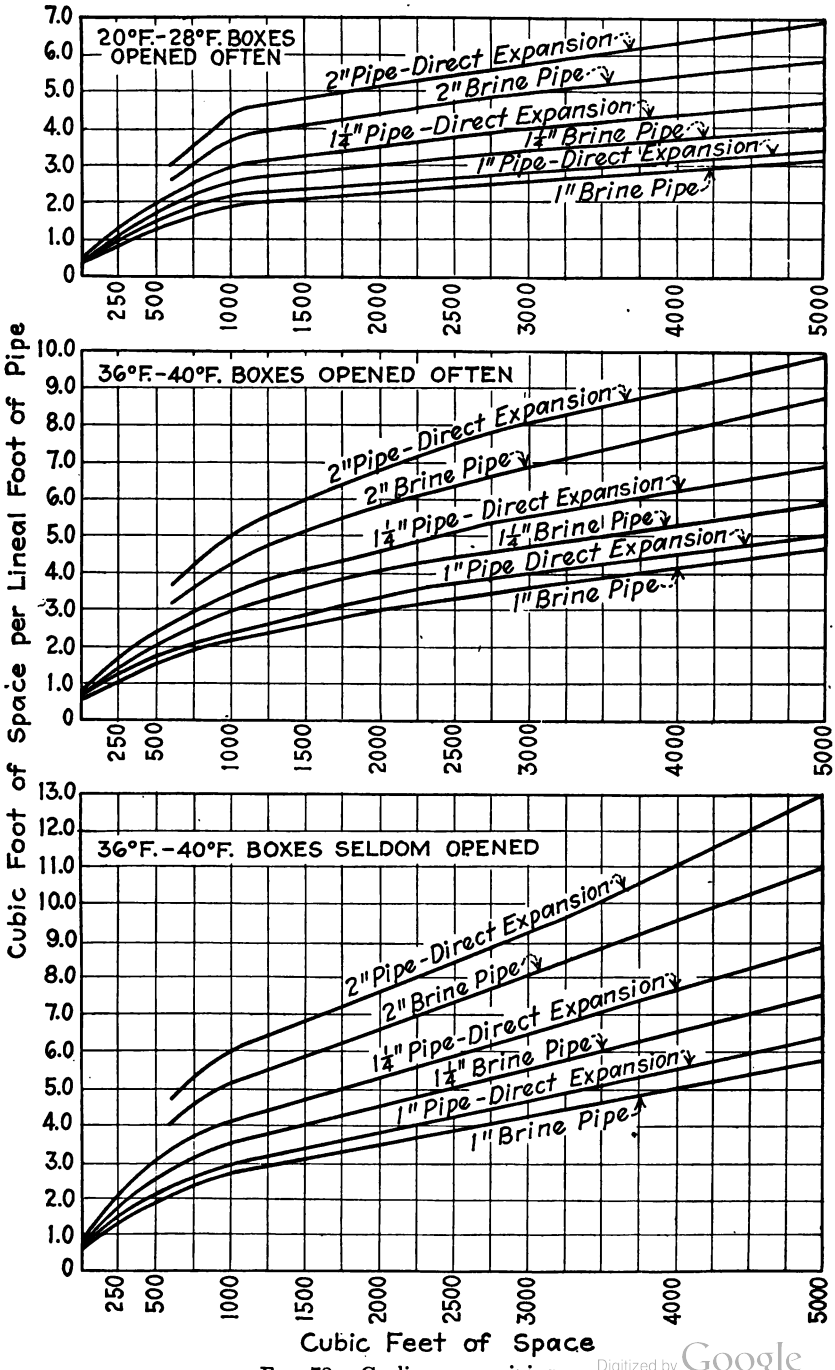


FIG. 78.—Cooling room piping.

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CAPACITY

Refrigerating capacity, tons	5	10	20	35	50	75	100	150	220	300	500
Ice capacity, tons.....	2.5	5	12	20	30	45	60	90	130	180	300

TABLE X.—CAN SIZES, WEIGHT OF ICE AND FREEZING TIME

Can size, inches	Weight of blocks, pound	Freezing time, hours	
		15 deg. F. brine	18 deg. F. brine
6 × 12 × 26	56	15	20
8 × 16 × 32	110	30	36
8 × 16 × 42	165	30	36
11 × 22 × 32	220	50	60
11 × 22 × 44	315	50	60
11 × 22 × 57	415	50	60

Table X covers the freezing time for a standard 300-lb. cake as well as the number of cans to allow per ton of ice for different brine temperatures.

Space for Storing Ice.—At least 50 cu. ft. of space per ton of ice should be provided.

Weight of Ice.—One cubic foot of ice weighs 57.5 lb. One cubic foot of water when frozen at 32 deg. occupies 1.0855 cu. ft.

Kilowatts per Ton of Ice.—The kilowatt-hours per ton of ice varies from 46 to 70 depending upon the capacity of the plant.

Ice per Ton of Coal.—Existing plants range from 2.25 to 9 tons of ice per ton of coal.

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TABLE XI.—COLD STORAGE TEMPERATURES

Articles	Degrees Fahrenheit	Articles	Degrees Fahrenheit
Fruit		*Liquids*	
Apples	32–36	Beer, ale, porter, etc.	33
Bananas	34	Cider	30
Berries, fresh	36	Ginger ale	36
Cranberries	33–36	Wines	40–45
Cantaloupes	40	*Flour and Meal*	
Dates, figs, etc.	50–55	Buckwheat flour	36–40
Fruits, dried	35–40	Corn meal	36–40
Grapes	34–36	Oat meal	36–40
Lemons	33–36	Wheat flour	36–40
Oranges	34–36	*Vegetables*	
Peaches	34–36	Asparagus	34–35
Pears, watermelons	34–36	Cabbage	34–35
Meats		Carrots	34–35
Brined	38	Celery	34–35
Beef, fresh	33	Dried beans	32–40
Beef, dried	36–40	Dried corn	35
Calves	32–33	Dried peas	35–40
Hams, ribs, shoulders (not brined)	20	Onions	36
Hogs	29–32	Parsnips	34–36
Lard	38	Potatoes	36.40
Livers	20–30	Sauerkraut	35
Sheep, lambs	32	*Miscellaneous*	
Ox-tails	30	Cigars, tobacco	35
Sausage casings	20	Furs, woolens, etc.	35
Tenderloin, butts, etc.	32	Honey	45
Fish		Hops	40
Fresh fish	20	Maple syrup, sugar	40–45
Dried fish	36	Oils	35
Oysters in shell	30–35	Poultry, dressed, iced	28–30
Oysters in tubs	25	Poultry, dry picked	26–28
Canned Goods		Poultry, scalded	20
Sardines	35–40	Game, to freeze	15–18
Fruits	35–40	Game, after frozen	25–28
Meats	35–40	Poultry, to freeze	15–18
Butter, Eggs, Etc.		Poultry, after frozen	25–28
Butter	18–20	Nuts, in shell	35–40
Butterine	18–20	Chestnuts	33
Cheese	34		
Eggs	31		

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TABLE XII.—SPECIFIC AND LATENT HEAT OF VARIOUS FOOD PRODUCTS

Substance	Composition		Specific heat above freezing in heat units	Specific heat below freezing in heat units	Latent heat of freezing in heat units
	Water	Solids			
Lean beef	72.00	28.00	0.77	0.41	102
Fat beef	51.00	49.00	0.60	0.34	72
Veal	63.00	37.00	0.70	0.39	90
Fat pork	39.00	61.00	0.51	0.30	55
Eggs	70.00	30.00	0.76	0.40	100
Potatoes	74.00	26.00	0.80	0.42	105
Cabbage	91.00	9.00	0.93	0.48	129
Carrots	83.00	17.00	0.87	0.45	118
Milk	87.50	12.50	0.90	0.47	124
Oysters	80.38	19.62	0.84	0.44	114
Whitefish	78.00	22.00	0.82	0.43	111
Eels	62.07	37.93	0.69	0.38	88
Lobster	76.62	23.38	0.81	0.42	108
Pigeon	72.40	27.60	0.78	0.41	102
Chicken	73.70	26.30	0.80	0.42	105

The figures in the last column showing the latent heat of freezing, have been obtained by multiplying the latent heat of freezing water, which is 142 heat units, by the per cent of water contained in the different materials considered, for as the solid constituents remain in their original condition, only the liquid or watery portion of these materials are concerned in the solidification or freezing of them.

THERMOMETERS FOR COLD-STORAGE ROOMS

To install an angle thermometer to enable the temperature inside a cooled room to be read from the outside, place the thermometer so that only the metal attachment is at the outer side of the cold-storage room. The sleeve of the thermometer passage should be filled with insulating material so that there will be no possibility of heat being transmitted to the thermometer bulb from the outside. Put a glass front over the recess in the wall containing the thermometer.

To absorb the moisture in a refrigerated room, use broken calcuim chloride spread out on a rack and provided with a drain for the brine formed, which should drain to a pail and be thrown away. Spread the calcium on a screen, if possible, to secure better air circulation around it. If the brine formed can be dripped over refrigerating coils, do so; the advantages are obvious.

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CHAPTER VI

INSULATION

In general, it may be said that if two adjacent bodies have different temperatures, the tendency is for an average temperature to be formed. For example, the hotter substance tends to become cooler and the colder one to become warmer. The method by which this may be accomplished may be by radiation, convection or by conduction. *Radiation* is the method by which the sun's rays reach the earth or by which one receives the sensation of heat from an intense fire perhaps some distance away. *Convection* is the method by which heat is conveyed by a moving substance, like hot-air or hot-water heating in the factory or house. Finally, *conduction* is the manner in which heat is carried along by the particles (molecules) of a solid, as in the case of a hot slicing bar. In the case of refrigeration the heat comes from the atmospheric air as a rule, and passes into the rooms which are refrigerated or the insulated tanks or pipes.

The problem of insulation is one of securing a substance that is practical as regards price, ease of application and long life under working conditions, one that will decrease the losses to a minimum, remembering that the initial cost of the insulation varies directly with the thickness of the material, whereas the losses due to lack of more or less material are a maximum of the summer months only.

From the foregoing it may be seen that the problem narrows down to the following: To conserve refrigeration, insulation of a suitable material must be provided on all sides of the cold-storage room and on the brine and return ammonia-pipe lines. Also that perfect insulation is impossible and that the proper amount and the kind of material to be used are dependent on conditions that prevail. These conditions vary with the aver-

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age summer temperatures, with the kind of cold storage or refrigerating duty performed (and the length of this service) and the cost of a ton of refrigeration.

THE CHOICE OF AN INSULATING MATERIAL

The insulation material is really a part of the building construction, and it must therefore be on a par with the rest of the building. It must be capable of withstanding deterioration or rotting, and, in addition, should be free of odor of any sort. It should be capable of maintaining itself in position and not settle in places to leave pockets or air spaces. It must be waterproof so that moisture may not collect, thus tending to mold and to diminish its ability to insulate, and it must be vermin-proof. Finally, the insulator must be of a high degree of perfection and must be capable of reducing the heat losses.

The problem of insulation is much more difficult so far as moisture is concerned than is the case with steam. With steam the tendency is to dry out the covering and to maintain this dryness as long as the temperature of the steam is maintained inside the boiler or vessel. In refrigeration the opposite is true and moisture has a tendency to travel toward the coldest surfaces and to keep collecting as time elapses. In this case also it is possible to have ice form, which may loosen or disintegrate the insulation.

THEORY OF INSULATION

The theory of insulation is somewhat difficult to understand at first. According to this theory there is no material that resists heat perfectly. Remember that insulating for refrigeration purposes is preventing heat from coming inward from the atmosphere to the region maintained at a temperature which is *below* its surroundings. However, it is found that air pockets of extremely small size are an insulation, and any material which has the greatest number of these pockets is the best insulations but each pocket must be separated from its neighbor. If the pockets were connected, then convection could

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take place as in the case of a hot-air furnace. From this it would naturally follow that the material that is the best insulator is also the *lightest* in point of *weight*, as a large amount of its volume is air. This is the reason that substances like mineral and animal wool, corkboard, balsa wood, shavings and sawdust have insulation values.

The use of air spaces naturally follows from what has just been said. However, it can be proved that large air spaces have little value. Therefore, a partition made up of alternate courses of lumber and air spaced is uneconomical of building space.

Sawdust and shavings were used extensively at one time and still have certain advantages. They are not waterproof nor are they vermin- or fire-proof. They lend themselves to cheap temporary construction, such as ice storage buildings or wooden buildings put up for a short life or for short refrigerating periods. The work should be carefully performed, and the shavings or sawdust should be carefully packed and a waterproof paper very carefully secured in place. Sawdust is likely to settle and leave a space in the upper part of the wall, so shavings, dried before using, are preferred.

Lately, for second-class construction, a large number of special materials have appeared in the form of convenient-sized boards. These are compressed wood and vegetable fibers so devised that the board may be erected like lumber and nailed into place. Being compressed, it has the characteristics of shavings and has the advantage of more efficient insulation and being able to retain its position. It is not vermin-, fire- or water-proof, and so it would generally be used only in wooden constructions.

Some success has been achieved in the use of mineral wool, the product of blast-furnace slag or of special mineral earths. These are made up in the form of boards and attempts are made to waterproof the material in the process of manufacture. When so done, it is easily erected in place, is fireproof and a good insulator. It is used in permanent construction and should maintain itself for years.

By far the most successful and popular insulating material

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is cork, obtained from the bark of trees found principally in Spain and Portugal. This bark has the air-cell construction which is necessary for the use we desire of it. For refrigeration use is made of the wastage in the manufacture of cork stoppers, etc. The cork particles are compressed and heated so that they adhere together and are made up into boards of convenient sizes or in forms to fit standard pipes or tanks. The cork is inherently waterproof, and in erecting the joints and courses are made up in asphalt pitch. In building construction there is also a final coat, as a rule, of cement plaster, which protects the corkboard and gives a smooth surface for suitable cleaning.

Special kinds of work also use granulated or pulverized cork, as in insulating the sides of ice-making tanks or the ends of cylindrical tanks. Usually, granulated cork should be used only where access is readily obtained to repack from time to time should settlement occur.

INSULATION VALUES OF BUILDING MATERIALS

Lately, numerous attempts have been made to find a suitable and correct method of figuring the insulating value of building materials. Elaborate experiments have been made, both with simple materials and with built-up construction. In these experiments it has been shown that there is on all material a "skin effect" which increases the insulating value. However, this is a matter that affects only the outside material, which for good insulation is usually cement mortar, or otherwise, perhaps, tongued-and-grooved lumber or brick. The real insulating material is neither of these.

Neglecting the skin effect, the insulating value of the material used is dependent on its conductivity. By this is meant the number of heat units that will pass through it, usually rated on each square foot, one inch thick, per hour or per 24 hr. with a difference in temperature on each side of the insulation of 1 deg. F. If this "leakage" is known for 1 deg. difference, then the leakage loss will be proportionately greater for 25, 50, 75 or 100 deg. F. difference in temperature.

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Likewise, if a material 1 in. thick will give a certain leakage loss, then one 2 or 6 in. will give proportionately less loss. This can be expressed in the formula:

$$Q = A \times K \times (t_{outside} - t_{inside})$$

where A equals area of wall or pipe surface in square feet, K equals conductivity, and t equals temperature in deg. F.

The value of K is obtained from tables of experimental values, as given below, or if use is made of several substances by the formula,

$$K = \frac{I}{\frac{T_1}{C_1} + \frac{T_2}{C_2} + \frac{T_3}{C_3} + ETC.}$$

where T_1, T_2, etc., equal thickness of each material in inches, and C_1, C_2, etc., represent conductivity as given in the tables for standard conditions.

Calculations: As an example of the use of the formulas, suppose a wall composed of 8 in. of brick, 2 in. board and ½ in. mortar is exposed to temperatures of 75 deg. and zero deg. F. on the two sides and it is desired to find the leakage per twenty-four hours for a total surface of 700 sq. ft.

The value of K becomes (see values in table) XIII

$$K = \frac{1}{\frac{8}{4.66} + \frac{2}{0.31} + \frac{\frac{1}{2}}{0.63}} = 0.111 \text{ B.t.u. per hour}$$

and the leakage is

$$Q = 700 \times 0.111 \times (70 - 0) = 5{,}400 \text{ B.t.u. per hour.}$$

Whereas the skin effect would tend to reduce the value of K and Q, yet the factor which is important is the perfection of the erection, which is subject to wide variations. Unless the joints are carefully made and air-tight, moisture will get in and freeze, causing the ultimate deterioration of the insulation. This is probably the factor most responsible for the fact that the laboratory results and the practical results do not agree, and we are justified in adding a liberal factor of safety to insure good operating results. For these reasons and for

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simplicity of calculation it is recommended that use be made of the method of calculation as given.

As regards the maximum temperature to be used in the calculation of leakage losses, it is customary to assume an average maximum temperature during the period of peak refrigerating loads. Some idea of average temperatures in the United States may be obtained from United States Government reports. It is not recommended that the highest local temperatures be taken, but the average temperature experienced during the 24 hr. for conditions that are likely to prevail for a period of days or a week or more. Of course the actual choice of the thickness of insulation for any particular case will be decided by the relative costs of the insulation itself and the book value of a ton of refrigeration. If the time of operation, days per year, is small, then one cannot economically install heavy first-cost construction. It would be too much like spending two dollars to save one dollar.

In conclusion, it is worth while putting special stress on the fact that the insulation must be kept dry, it must be kept in place and should be odorless. It should be easily placed in position and be of a material of economical insulating value.

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TABLE XIIa.—REPORT OF TESTS CONDUCTED BY THE UNITED STATES BUREAU OF STANDARDS, WASHINGTON, D. C., COVERING THERMAL CONDUCTIVITIES OF VARIOUS INSULATING MATERIALS

Material	Conductivity 24-hr. ft.-sq. in. thick degrees Fahrenheit	Density	Nature of material
Air	4.0	.....	Horiz. layer heated from above radiation.
Colorax	5.3	0.064	Fluffy finely divided mineral matter (elim.).
Hair felt	5.9	0.27	
Keystone hair	6.5	0.30	Hair felt confined between layers building paper.
Pure wool	5.8	0.107	Firmly packed.
Pure wool	5.8	0.102	Firmly packed.
Pure wool	6.3	0.061	Loosely packed.
Pure wool	7.0	0.039	Very loosely packed.
Cotton wool	7.0	0.000	Firmly packed.
Insulite	7.1	0.19	Pressed wood pulp—rigid, fairly strong.
Linofelt	7.2	0.18	Vegetable fiber confined between layers paper—soft and flexible
Corkboard (pure)	7.4	0.18	
Eelgrass	7.7	0.25	Inclosed in burlap.
Flaxlinum	7.9	0.18	Vegetable fibers—firm and flexible
Fibrofelt	7.9	0.18	Vegetable fibers—firm and flexible
Rock cork	8.3	0.33	Prest rock wool with binder—rigid.
Balsa wood	8.3	0.12	Very light and soft.
Waterproof lith	9.8	0.27	Rock wool, vegetable fiber and binder, not flexed.
Pulp board	10.4	.....	Stiff pasteboard.
Air cell ½-in	10.7	0.14	Corrugated asbestos paper, inclosing air spaces.
Air cell 1-in	11.5	0.14	Corrugated asbestos paper, inclosing air spaces
Asbestos paper	11.8	0.50	Fairly firm but easily broken.
Infusorial earth	13.9	0.69	(Block).
Fire felt (sheet)	14.3	0.42	Asbestos sheet coated with cement—rigid.
Fire felt (roll)	15.3	0.68	Soft, flexible asbestos.
3-ply Regal roofing	16.7	0.88	Flexible tar roofing.
Asbestos mill board	20.2	0.97	Prest asbestos, fairly firm, easily broken.
Woods—kiln dried			
Cypress	16.0	0.46	
White pine	19.0	0.50	
Mahogany	22.0	0.55	
Virginia pine	23.0	0.55	
Oak	24.0	0.61	
Hard maple	27.0	0.71	
Asbestos wood (sanded)	65.0	1.97	Asbestos and cement—very hard and rigid.

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INSULATION

Insulation.—Heat transmission per square foot per 24 hr. per degree Fahrenheit of mean temperature difference, "J. M." pure cork.

Thickness, inches	B.t.u.
1	6.40
1.5	4.25
2	3.20
3	2.15

Brick Walls (solid).—B.t.u. per square foot per degree Fahrenheit of mean temperature difference inside and outside for 24 hr.

Thickness of wall, inches	B.t.u.
4	5.5
9	4.5
14	3.6
27	2.6
36	2.2

Stone Walls (solid).—

Thickness, inches	B.t.u.
6	6.2
12	5.5
24	4.5
36	4.1

Windows.—B.t.u. per square foot per 24 hr. per degree Fahrenheit mean temperature difference inside and outside, 12; double windows, 7. Single skylight, 26.5; double, 12.

Other Materials.—Same basis as above, except materials are 1 ft. thick.

Sawdust	1.1
Soft paper felt	0.5
Granulated cork	1.3
Pine wood	2.0

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TABLE XIV.—TRANSMISSION OF HEAT THROUGH 1½- TO 2-IN. IRON PIPES PER SQUARE FOOT PER HOUR PER DEGREE OF DIFFERENCE IN TEMPERATURE

Mode of Operation	B.t.u.	Example
Ammonia gas inside, water outside	50	*Submerged condenser.*
Ammonia gas inside, running water outside	60	*Atmospheric condenser.*
Ammonia gas inside, brine outside	25	*Brine tank.*
Ammonia gas inside, wort outside (counter current)	60	*Dir. exp. beer cooler.*
Ammonia gas inside, air outside	2–8	*Direct expansion.*
Cold brine inside, water outside	80	*Water cooler.*
Cold brine inside, water outside	60	*Distilled—water cooler.*
Cold brine inside, wort outside	70	*Brine beer cooler.*
Cold brine inside, wort outside (counter current)	75	*Baudelot cooler with brine.*
Ammonia liquor inside, water outside (counter current)	60	*Absorber.*
Ammonia liquor inside and outside (counter current)	50	*Exchanger.*
Water inside and outside (counter current)	50	*Exchanger.*
Steam inside, water outside (counter current)	500	*Steam condenser.*
Steam inside, water or ammonia liquor outside	300	*Ammonia liquor still.*
Steam inside, air outside	2–3	*Steam pipes.*

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TABLE XV.—TRANSMISSION OF HEAT THROUGH VARIOUS INSULATIONS PER SQUARE FOOT IN 24 HR. PER DEGREE OF DIFFERENCE IN TEMPERATURE

	B.t.u
2 boards with paper, 1-in. air space, 5-in. Nonpareil sheet cork,	0.9
paper, board	0.9
1 board with paper 3-in. Nonpareil sheet cork, paper, board	2.1
1 board with paper, 2-in. Nonpareil sheet cork, 2 boards with paper	3
2 boards with paper, 4-in. granulated cork, 2 boards with paper	1.7
1 board, 2½-in. mineral wool, paper, board	3.62
1 board, paper, 1-in. mineral wool, paper, board	4.6
2 boards with paper, 8-in. mill shavings, paper, 2 with paper dry	1.35
Same, damp	2.1
1 board, 2-in. air space, board, 2 in. "Lith," paper, board	1.8
4 boards, 1-in. flax sheet lining, 2 papers	2.3
1 board, 6-in. silicated strawboard (air cell), layer of cement	2.5
4 boards, 4 quilts of hair	2.52
2 double boards with 2 papers, 1-in. hair felt	3.32
1 board, paper, 2-in. calcined pumice, paper, board	3.4
1 board, 2-in. pitch, board	4.25
4 double boards with paper (8 boards) and three ⅜-in. air spaces.	2.7
2 double boards with paper (4 boards) and 1-in. air space	3.71
4 boards with 2 papers, solid, no air space	4.28
Brickwall, 3-in., hollow tile, 4-in. mineral wool, 3-in. hollow tile,	
cement plaster	0.7
Concrete floor, 3-in. book tiles, 6-in. dry underpiling, double space	
hollow tile arches, cement plaster	0.8

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CHAPTER VII

OPERATION OF THE COMPRESSION SYSTEM. IN THIS CHAPTER APPEARS A NUMBER OF PRACTICAL ARTICLES OF SUCH INTEREST THAT THE AUTHOR IS CREDITED WITH THE MATERIAL

REMOVING THE AMMONIA CHARGE

BY E. W. MILLER

The greatest trouble in removing a charge in the ordinary way is that the ammonia liquid is very slow in coming down from the condenser. It seems to like to hang up there. I first made a connection to the liquid receiver from which the liquid ammonia was drawn off into empty drums as fast as it came down. The drums were placed on a scale so we could tell when they had the proper amount in them. When ready to start, I shut off the valve in the liquid line to the expansion coils on the ice tanks and pumped as high a vacuum as possible on the tanks. The liquid ammonia, as fast as it came down into the receiver, was drawn off into the drums.

Let Air into the Steam.—After I had pumped down until about 25-in. vacuum could be maintained with the machine barely turning over, only about half of the charge in the system was in the drums. The rest was coming down into the receiver very slowly. The condenser was an extra large one for the work it had to do, which made matters that much worse. When the expansion coils were apparently pumped empty, as indicated by the slow speed of the machine maintaining the vacuum, I loosened one of the flanges in the liquid line above the stop valve at the receiver and let air into this part of the system. Before I did this, I shut off the expansion valves on the tanks on all coils but one. On this coil I opened the expansion valve wide.. In this way the air rushing in through the opened flange carried the dead liquor in the liquid line and in

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the coil ahead of it back to the compressor. After the first coil had blown for about 15 min., the expansion valve on this coil was closed and the next opened.

Expansion Coils Quickly Cleaned.—This procedure was followed until every coil had been blown. Then the first one was given another short blow, the expansion and suction valve closed to keep anything from backing up into it from the other coils, and the same thing was done with the other coils. In this way the expansion coils were well cleaned. This was not the only point gained. My reason for letting the air into the system was to fill the condenser with air and thus increase the condenser pressure, when I concluded the ammonia in the condenser would all condense and come down more readily. Before I opened the flange in the liquid line to let the air into the system, I went up on the condenser and shut off the liquid valve on each stand but one. In this way the flow down to the receiver was all through the one coil. As soon as this was done, the liquid in this coil came down in a rush, as shown by the gage glass on the receiver. Soon no more came down, so I went up and shut off the liquid valve on this stand to prevent liquid from backing up into it from the other coils when the liquid valves on these were opened. Then I opened the liquid valve on the next stand, when the liquid came down quickly, as it had done with the first one. This performance was followed with each condenser stand. The condenser pressure was maintained at 210 lb. by varying the speed of the machine, and with 60-deg. condenser water there was nothing to do for the ammonia but condense, and with only one coil open at a time to the receiver, the flow into the coil forced the liquid down ahead of it. It was a simple matter to tell when a coil was clear by observing the gage glass on the liquid receiver. After I had gone over the entire condenser in this manner once, I went over it the second time, but got little ammonia.

One Coil Purged at a Time.—The expansion coils were each provided with a purge valve at the bottom and were fed from the top. As soon as all the liquid was out, the joint in the liquid line was closed and a joint in the suction header of the machine opened. In this way air was pumped through the

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condenser, liquid receiver and liquid line into the expansion coils. The purge valve on one coil at a time was opened and each coil blown out thoroughly. There was little ammonia in any of them as the first blowing had cleared them quite well.

In addition to getting the charge out in a hurry, nearly all the dead liquor and water was cleaned out by circulating air through the system in this manner. After all the coils had been blown, the liquid receiver was blown. Then everything was closed and as high a vacuum as possible pumped on the whole system by closing the discharge stop valve on the compressor and blowing the air out through the purge valve on the machine. Then the new charge was put in. As an indication of the thoroughness with which the system was cleaned out, a test was made of the ammonia after it had been working a week, and only a slight trace of moisture was found. After the new charge was put in, the capacity of the plant was increased nearly 30 per cent, so it did not take long to make up for the time lost in making the change. The engineer should be careful about employing the method described, as there is always the possibility of an explosion where an air pressure is created in any part of an ammonia system while the charge or a part of the charge is present. This applies particularly to the condensing, or high-pressure, side of a refrigerating plant.

There is also the question of danger due to the presence of oil in the high-pressure oil trap and connections, which, when subjected to high air pressures and consequent temperatures, sometimes creates a condition that results in severe explosions.

LOCATING AMMONIA LEAKS IN THE REFRIGERATING SYSTEM

One of the most troublesome problems that confronts the refrigerating engineer at times is to account for the loss of the ammonia charge. He knows from the action of the system and by the gage glass on the receiver that it is gone; but to account for it is difficult.

The greatest single source of loss is leaks. It is safe to say

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that if every plant in the country was tight we would not hear so much about deterioration of ammonia, and the question of loss of charge would arise but seldom.

Where the condenser is purged frequently, this will occasion considerable loss if it is done in the generally accepted way. As far as deterioration is concerned, if the system is kept reasonably free from oil it is not likely that this will cause an appreciable loss. Where there is a large amount of oil in the system and the compressor is run very hot, it seems that the ammonia decomposes more or less; and this of course will mean a loss. Generally, however, it is the undetected leaks that cause the greatest loss, and every engineer should make it his business to be on the lookout for unsuspected leaks all the time and attend to them as soon as they are discovered.

The biggest job is to locate the leaks; after they are once found, the remedy is usually quite simple. Often it is only a case of drawing up on the bolts in the flanges. For locating the leaks there is nothing better than the old sulphur stick, unless the leak happens to be under water or brine. In this case use litmus paper or Nessler's solution.

Sulphur sticks can be made by anyone, with little trouble and expense, and no refrigerating plant should be without a good supply of them. Sulphur can be had at any drug or chemical store and costs but little. To make the sulphur sticks, place the sulphur in a tin can or pan (a bread pan is just the thing) and place it on anything that will heat it to about 250 deg. F. A hot steam pipe or the boiler breeching is a good place to do this. The bare part at the end of the cylinder of many Corliss engines above the valves is the most convenient place. The top of the steam chest on a pump will serve if the bolt heads do not keep the pan from coming in contact with the hot metal. Melting the sulphur over a fire or a forge is not satisfactory, because the sulphur generally takes fire. If it is desired to melt it in a forge or over a blow torch, it is best to put a plate of iron over the flame and then place the pan with the sulphur in it on top of this plate. In any case it is good practice to cover the pan containing the sulphur, to prevent radiation.

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While the sulphur is melting, get some candle-wick and twist it into rope about four strands thick. Cut it up into lengths about a foot long. When the sulphur is melted, dip the candle-wick into it and let it soak for a minute or two and the job is done. Care must be taken not to get the molten sulphur on the hands or fingers, as it hardens instantly and produces painful burns. If candle-wick is not available, get a pine board and make sticks about a foot long and dip these into the sulphur in place of the candle-wick. Either serves very well. If you

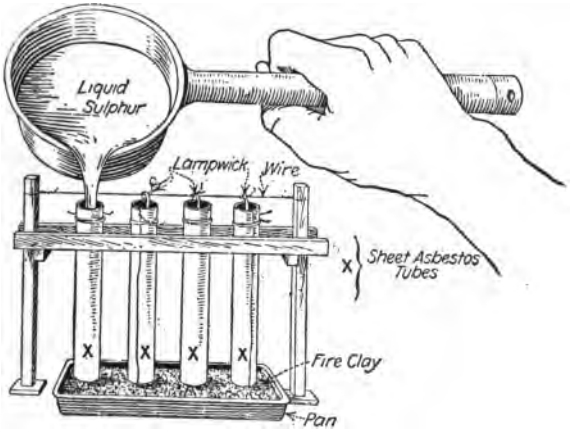


FIG. 79.—Making sulphur candles.

wish to make considerable smudge, get some sheet asbestos and roll it up into tubes. Put a little fireclay in the bottom to plug the end of the tube, and pour the tube full of sulphur (see Fig. 79).

To test for leaks, light one of the sulphur sticks and hold it up to the suspected point, under it if possible. The fumes of burning sulphur are practically invisible; but if they get mixed with ammonia fumes, a dense grayish-white smoke is formed (see Fig. 80).

Leaks in water or brine are most readily detected with litmus-paper or Nessler's solution. Litmus paper can be had at any drug store or from the ammonia manufacturer. To use this paper, simply dip it into the suspected water or brine.

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If there is ammonia present, the paper will turn blue. Small traces of ammonia will barely color the paper, while any large proportion will turn it a deep blue. Litmus paper, however, will not give reliable indications in calcium brine. Calcium brine turns litmus paper blue just as ammonia will, and for this reason it is useless for detecting ammonia in this kind of brine. Instead, use Nessler's solution. It is also just as good as litmus paper, if not better, for salt-brine and water tests.

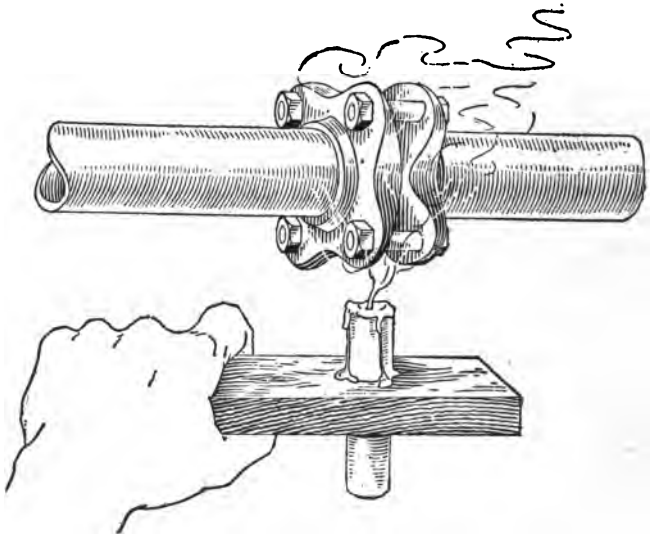


FIG. 80.—Detecting ammonia leaks.

The formula for Nessler's solution is as follows: Dissolve 17 grams of mercuric chloride in about 300 c.c. of distilled water; dissolve 35 grams of potassium iodide in 100 c.c. of water. Add the former solution to the latter, with constant stirring, until a slight permanent red precipitate is formed. Next dissolve 120 grams of potassium hydrate in about 200 c.c. of water; allow the solution to cool and then add it to the previous solution and make up with water to one liter. Add mercuric chloride solution until a permanent precipitate again forms. Allow to stand till settled and decant off the clear

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solution for use. Put it in glass-stoppered blue bottles and set away in a dark place.

In calcium brine, Nessler's solution will always form a precipitate; but if no ammonia is present, the precipitate will be almost white. In water or brine the precipitate will be yellow if there is but a trace of ammonia present, and a reddish-brown if there is considerable ammonia in the sample. To make the test, get a sample of the suspected water or brine in any kind of glass vessel and put a few drops of the solution into it. If it changes color, as stated before, the brine or water contains ammonia and preparations should be made to locate the leak.

To get the leaks as soon as they occur, tests should be made systematically at regular intervals. Go over all the open joints with a sulphur stick and test the jacket water, condensing water and brine with litmus paper or Nessler's solution. Most of the leaks will generally be found around the compressor, discharge line and condenser. This is partly due to the high pressure and partly to the temperature changes at different times on this side of the system. A double-pipe condenser especially will need close watching for leaks after it gets old.

If the bolts in the flanges are loose, the leak may be stopped by drawing them up tight. Never try to stop an ammonia leak—on the high-pressure side of the system especially—by drawing the nuts up on the bolts by hammering on the end of the wrench after the nuts are drawn as tight as they will go by hand. If the joints are drawn up as tight as they will go by hand, the safest thing is to pump out and put in a new gasket. It is never advisable to hammer the nuts at a leaky joint as a fatal accident might follow.

In breaking open joints, even if one is positive that everything is pumped out, it is always best to go slow and be careful. Loosen up the bolts on the flange a little and then gently pry the joint apart. If there should still be pressure on the joint, it is then a simple thing to draw up the bolts again and save the ammonia, and possibly your life. There have been cases where men lost their lives from disregarding

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this precaution, the men assuming the pressure to be off because the system had just been pumped down. Considerable pressure may accumulate on the line, and men be caught in a corner where they cannot get out.

Now when the joint is opened and the old gasket is out, the next question is what kind of gasket material to use. For the high-pressure side of the compressor, the discharge line and the condenser many prefer the good-grade sheet-asbestos packings. For the liquid line and the low-pressure side of the system, including the expansion coils, lead is usually preferred. Lead does not give good service on the high-pressure side, because every time the compressor and the discharge line get hot, the joint expands and the lead squeezes together; and when the line and joints contract, the joint gets loose and starts to leak. There is no elasticity to lead, while the asbestos-sheet packings are elastic. For this reason the high-pressure joints, if they are provided with lead gaskets, need constant tightening if the temperature of the machine varies. Also a lead joint, if it starts to blow, cuts the lead out and it is nearly impossible to stop it afterward by drawing up on the bolts. If an asbestos gasket is covered with a light coating of heavy cylinder oil and then as much graphite as will stick is dusted on and pressed down, not rubbed in, with the tips of the fingers, the joint can be opened any number of times with ease and the gasket will last indefinitely.

A PUZZLING AMMONIA LEAK

By E. W. Miller

The foreman of a new department complained about the odor of ammonia in one of the rooms. A man sent to investigate reported back that the odor was there, but that he was unable to find its source. It was required to maintain a temperature of 40 deg. F. in the room, and for this purpose two direct-expansion coils were hung on the two side walls. Compressed air at high pressure was used for operating two industrial machines in this room; the rest were driven by electric motors.

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Arming myself with a good supply of sulphur sticks, I proceeded to the department with the intention of immediately finding and stopping the leak. I failed. Smoke from the sulphur sticks filled the room, and everybody had to leave.

Sunday afternoon I could not smell the slightest trace of ammonia, and tests with the sulphur stick were fruitless. Both expansion coils were working. Deciding that the leak had taken up overnight, I let it go at that. Monday forenoon the foreman complained again about the ammonia, and he gave me a "piece of his mind" for not having attended to the leak the day before. When I got there about 9 p. m. the smell was nearly gone. Tests with the sulphur stick failed to show a leak at any particular point, but the smoke all over the room indicated that the whole atmosphere contained ammonia. The engineer who had charge of handling the expansion valves in this room informed me that the smell disappeared entirely toward morning. The watchman told the same story, and in disgust I went home.

A few days later we had occasion to use the air hose in the power house for cleaning the armature of a small motor. I happened to be manipulating the hose. The air blew back in my face, and I got a strong whiff of ammonia. Investigation disclosed that the air from the hose was the source of the smell. Going to the compressor supplying the air and opening the indicator cocks, I found ammonia-laden air coming from them. Evidently the air supply to the air compressor was mixed with ammonia.

This compressor was used mainly for blowing a well. Another compressor supplied the greater part of the plant with air at a much lower pressure. The only department supplied with the high-pressure air was the one where we had trouble with the ammonia odor.

The air for this compressor was drawn in through a sheet-iron pipe which ran up along one wall of the engine room nearly to the roof. Directly across the top of this intake of the compressor was one of the liquid lines from the ammonia condenser, and one of the joints, a leaky one, on the line was

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directly over the air intake to the air compressor. Drawing up on the bolts of the flange of the joint at once remedied the trouble.

CLEANING THE AMMONIA CHARGE AT THE END OF THE SEASON

By F. W. Frerichs

Overhauling should be done at reasonable intervals, say once a year, and consists in cleaning the ammonia charge, cleaning the brine, cleaning all parts constituting the plant, repairing all defects in apparatus and machinery, testing all parts of the plant and recharging and getting ready for new work.

Any ammonia charge may be contaminated by dirt left in the system before charging or by contact with packing materials, gaskets and lubricants during operation. Gasoline and coal oil used for cleaning, soap or lard oil used for cutting threads on pipes, grease, wax or asphalt contained in packing and gaskets are among the materials that may impair the ammonia. Most frequently, however, a bad quality of compressor oil is responsible for deterioration of the ammonia charge. Liquid anhydrous ammonia is to some extent a solvent for many of these materials, some of which can be eliminated by redistilling the ammonia. This may be successfully done in the following manner before cleaning the machine:

Cleaning Ammonia from Compression Plant.—Toward the end of the season run the ammonia charge down to a small volume, and upon stopping work in the plant transfer all the ammonia into the refrigerating coils. If there is more ammonia than can be conveniently worked in the refrigerating coils, store some of it in ammonia-shipping cylinders, returning it later to the freezing coils for reworking. Then close the expansion valves and detach the receiver from the feed line, leaving a valve near the receiver. Remove from the compressor all oil, clean the discharge pipe, condenser and receiver thoroughly by blowing steam through them and dry them by air while they are still hot. By running the compressor slowly, you can distill off most of the ammonia from the freezing coils,

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compressing it at the same time and accumulating it in the receiver, from which you may withdraw it for storage into shipping cylinders. Care should be taken to run the compressor so slowly that only ammonia vapors and no liquid anhydrous ammonia and oil can enter the suction pipe of the machine.

A good way of drawing off anhydrous ammonia from the receiver into shipping cylinders is as follows: Place the empty cylinder in a horizontal position upon the platform of a scale to show the weight of ammonia removed, turning it so that the valve stem points downward. Connect with the valve on the receiver, which previously was connected with the expansion valves, and open the valves near the reciever and the cylinder, whereupon the pressure in the system will force a small quantity of ammonia into the cylinder. Then close the valve near the receiver, and by opening a valve attached to a tee in the connecting line, relieve the pressure in the cylinder, thereby removing most of the air and cooling the cylinder by evaporating some of the ammonia contained in it. Upon closing the valve near the tee and opening the valve near the receiver, ammonia will flow freely into the shipping cylinder. To prevent bursting at slightly increased temperature, care should be taken that the cylinders are not overloaded. Cylinders 12 in. in diameter by 7 ft. long carry safely 150 lb.; cylinders 10 in. by 7 ft., 105 lb.; cylinders 10 in. by 4 ft., 55 lb. In this manner all the ammonia contained in the machine may be withdrawn gradually in a purified condition, the compressor taking it from the freezing coils to the condenser and through the receiver to the storage cylinders. If this is done, the plant is empty and is ready for cleaning.

The advantage of redistilling ammonia before recharging it into the system manifests itself by regular running at increased capacity, which means less wages and a smaller fuel bill per ton of ice.

TESTS FOR FOREIGN GASES

Sometimes, in taking foreign gas out of a machine, there may be a question as to whether there is enough coming out to

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justify one in leaving the gas valve open. When such a question arises, take a pint or quart bottle with a large mouth (a milk bottle is handy for this purpose) and fill it with water. With the open end of the bottle submerged in a bucket of water invert it, and hold it in this position; and at the same time insert the end of the foreign gas hose in the end of the inverted

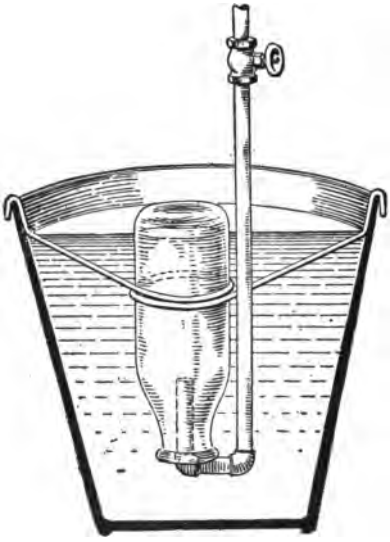


FIG. 81.—Tests for foreign gases.

bottle, Fig. 81. Some idea of the amount of air and non-condensable gases coming out may be obtained by taking note of the rapidity with which the bottle empties of water as it is displaced by air or gas. Ammonia gas would of course be absorbed by the water. One must not forget that time is required for such absorption.

CORROSION IN REFRIGERATING SYSTEMS

The following are concrete instances in which corrosion has actually taken place with resultant destruction to the positive metal. However, one should remember that as to proper combinations of metal in refrigerating system there can be no fixed rule.

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When brass valves are joined with wrought-iron nipples or pipe, the iron is attacked.

If cast-iron fittings are joined with wrought-iron nipples or pipe, the wrought iron is usually attacked more readily than the cast iron, although it too is eventually destroyed.

Where brass liners, etc., are in cast-iron brine pump, the iron is eventually destroyed.

When galvanized iron is joined with brass valves, the iron is rapidly destroyed, even though the iron has been regalvanized, for the zinc is even more strongly electro-positive toward brass than is iron.

In many cases, the use of galvanized-iron piping where there is much moisture and a free circulation of air has proved to be bad practice, especially if there are many pin holes in the zinc coating. The zone of protection afforded iron by zinc in galvanized iron is comparatively restricted, and, inasmuch as protection is given at the expense of the zinc, it is obvious that the more iron is exposed, the faster will be the destruction of zinc, zinc being electro-positive to iron.

If steel piping is joined with cast-iron fittings the steel piping or nipples will be first attacked, the cast iron also showing attack.

Steel piping when joined with brass valves or unions is vigorously attacked.

In copper vessels to which are attached iron or steel pipes or fittings the iron is attacked.

Iron or copper piping is often made up with red or white lead at the joint. As a rule, corrosion of the iron or copper may be expected after a time, especially in brine lines. Flange joints should always be used in preference to screw joints wherever practicable.

Corrosion will take place with aluminum when joined with copper or brass.

While it is quite impossible to give particular combinations of metals for use about refrigerating plants, the following seem to be good practice:

Malleable-iron fittings with wrought-iron piping or nipples as opposed to cast-iron fittings.

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Copper or brass valves or unions, etc., joined with iron but insulated by means of a heavy shellac or varnish used in making up the joints. Such means have proved successful in a number of instances as opposed to direct contact of copper or brass with iron.

Cast-iron fittings and cast-iron piping.

Where it is necessary to join unlike metals without insulation, it is good practice to use short fittings either side of the valve or second metal fitting. For example, in the case of a brass valve in iron-pipe lines it is well to use short iron nipples or flanges at each side of the valve so that when the iron is destroyed, it will be necessary to restore only these short pieces instead of the whole length of pipe.

In many cases it has been noted that iron galvanized by the sherardizing process may be successfully joined with ordinary iron or even with copper and brass, whereas iron galvanized by the hot or electrolytic process is usually destroyed under such conditions.

Steel pipe joined to copper or brass invariably sets up rapid corrosion.

GALVANIZED PIPE FOR AMMONIA

By William Schliemann

While in charge of a large cold-storage warehouse in an Ohio city, I had about 7,000 ft. of 2-in. direct-expansion piping to install, and to avoid rusting and odors due to painting, I determined to use galvanized pipe. I was not aware, nor were operating men with whom I talked, that ammonia had any deleterious action on galvanizing, so extra-heavy galvanized piping and the necessary valves and fittings were ordered. We had always had success in this plant, but soon trouble started. The head pressure went up to the limit and compelled shutting down, as it seemed that oceans of water would not keep the condenser pressure down. The condenser was of 125-ton capacity, counter current, though there were only 75 tons of compressor capacity in that part of the warehouse. We purged and purged, as the gases seemed to be coming constantly. Getting down to normal pressures, we started

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again only to find that we could get nothing but liquid to the compressor. Operating slowly and with the suction valve cracked, we finally got some kind of service. Suddenly up went the condenser pressure again, and more purging followed. We purged frequently night and day for a week. We managed to keep nearly normal temperature in the cooler, and we finally got down to regular operation again. I sifted the matter and finally blamed it on the galvanized piping.

I had not charged any ammonia into the system since putting in the galvanized coils, but had charged three days previously and had been operating all right.

The pipe and fittings were galvanized on the inside, and the ammonia probably dissolved the galvanizing and in doing so formed a gas. The system contained considerable oil, and this no doubt had contributed to the trouble. Finally the trouble ceased, evidently when the galvanizing was gone.

When taking down some old ammonia condensers, I found that where the hot gas entered the galvanizing had disappeared from the side of the pipes, but on the pipes subjected to moderate and cold temperatures it was practically intact. In the hot pipes the galvanizing had deposited in dust form in the bottom of the pipes.

In another plant I observed practically the same effect, but did not notice that galvanizing aided the deterioration of the ammonia.

LOSS OF AMMONIA

The chief cause of loss of ammonia is by leakage. Many claim that the ammonia deteriorates or wears out, but nearly all the ammonia makers protest that this is not so. The stuffing box is one of the greatest offenders. In most plants more than half of the ammonia wasted is lost around the stuffing box. The stems on all the valves that are much used generally leak. All the high-pressure joints should be watched, as they may start leaks any time; the condenser must also be watched. Regular inspection and test with the sulphur stick will keep the ammonia loss to a minimum if the stuffing box is kept in good order.

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Stuffing box troubles are generally caused by one of two things—bad rods or bad management. If a rod is scored, has shoulders on it or is worn more in the center than at the ends or is out of line, it should be attended to as soon as possible, as with the best of care it will not work well and it generally loses more ammonia than is represented by the cost of repairing it.

If the rod has been turned down, there should be a junk ring put in the bottom of the stuffing box, otherwise the packing will squeeze out between the rod and the bottom of the box.

On double-acting machines there is generally a gas-relief line running from the lantern in the center of the packing to the suction line; this sometimes gets blocked with bits of packing and foreign matter, and there is no outlet for the gas that leaks past the first section of the packing to get away, and this puts full pressure on the last, or front, section of the stuffing box, causing the packing to let the gas blow out.

Getting "Pocketed" Liquid Circulated.—Sometimes the ammonia will "hang up," or pocket, in some part of the system, and there will be all the signs of an insufficient charge, when there really is enough to go around if it were all in circulation. This generally happens where the charge is just about sufficient and no more. Where the expansion coils are fed at the bottom and there is a wide difference in the temperature of the different rooms, the ammonia will accumulate in the rooms where the coldest coils are located. Take for illustration a small packing house run with one machine: There are a couple of chill rooms where the temperature varies from 50 to 35 deg., one or two others kept at 35 deg., and the freezers, held at 15 deg. In the afternnoon and early evening, when the chill rooms contain much hot meat, these coils will boil off the ammonia faster and the back pressure will rise. Increasing the back pressure reduces the temperature difference between the room and the ammonia in the freezer coils, and consequently the rate of evaporation becomes slower; in fact, if the back pressure increases enough, the evaporation will stop altogether and the coils will fill with liquid. The only remedy when the system runs short of ammonia in this

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way is to shut off the low-temperature rooms and let the ammonia boil out, or run part of the time on the low-temperature rooms and part on the high. Care in alternating this way will soon get the ammonia into circulation.

The ammonia will accumulate in the condenser if the head pressure is lowered. Lowering the head pressure reduces the volume of gas in the condenser and increases the volume of liquid. So, if the ammonia runs low in the receiver, it can sometimes be brought down by raising the head pressure or, what amounts to the same thing, speeding up the machine.

REVIEW OF SOME AMMONIA EXPLOSIONS

By Chas. Bromley

For two reasons chiefly—because of the rapid development of the refrigeration art and because of the number of serious explosions of refrigerating apparatus—critical studies of the causes of such explosions are essential to the healthy growth of the industry. The number and kind of explosions are not alarming, and in that there is optimism for user, manufacturer and engineer. But there must be a more general recognition that, first, any high-pressure ammonia plant, like any boiler plant, is pregnant with potential danger; second, that ammonia in a refrigeration system may become a powerful explosive; third, that the employment of untrained men to care for such apparatus may properly be regarded as criminal.

The explosibility of gases generated in a refrigeration system has been well established and analyzed in recent technical literature by Prof. R. Plank and F. L. Fairbanks, and only the "high spots" of their investigations need be mentioned here. Professor Plank has shown that reactions bringing about decomposition of ammonia are rapid at temperatures prevailing in practice; he has shown that catalytic action in the compressor cylinder and discharge pipe is responsible for decomposition. The mechanical impact of sudden compression is an influence that is difficult to calculate but which is known to exist.

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Mr. Fairbanks has been unable to produce explosion of refrigeration-system gases by detonation, but has shown that the lubricating oil at operating temperatures becomes a hydrocarbon gas which, when subject to ignition, is highly inflammable when mixed with certain proportions of air and highly explosive when mixed with certain other proportions. Actual explosions corroborate this statement. Mr. Fairbanks' experience has also shown that when zinc or zinc alloys are in contact with ammonia at high temperatures, decomposition is accelerated and considerable hydrogen liberated. All experience shows that for an explosion of apparatus containing ammonia—or more correctly, the ammonia itself under operating conditions—there must first be a break liberating to the atmosphere the already-present gases of decomposition and the ammonia, and second an arc or flame to ignite these gases, which quickly unite with the oxygen of the air to form an explosive mixture. It is probable that after the explosion has started, decomposition occurs rapidly enough in the intense heat to keep up the supply of the explosive until it burns itself out. That high operating temperatures without the effect of impact will produce decomposition is shown by the fact that the gases issuing from the purge pipe of an absorption system may be ignited almost any time.

On the afternoon of Sept. 14, 1916, the receiver tank of a 20-ton refrigerating plant belonging to the Interstate Milk and Cream Co., Newark, N. J., exploded, instantly killing six men and seriously injuring four others. The plant was a new one on which the final tests were being made preparatory to turning it over to the owners. Without much of a report and doing comparatively little damage to the building, the receiver opened its entire length.

The receiver had been tested to 300-lb. air pressure, and about 3 weeks prior to the accident a preliminary test had been run on the plant, at which time everything was found to be working satisfactorily. The normal working pressure of this system is 125 lb.

The receiver was 20-in. diameter and 10 ft. long, with a $\frac{1}{16}$-in. shell, the seams autogenous-welded. From all appear-

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ances, the longitudinal seam failed about midway between the headers, and starting at this point, it opened the entire length. One header seam opened for about three-quarters of the circumference, the other two-thirds of the circumference.

It is believed that further investigation will show safety valves are a necessary part of a refrigerating plant and that this accident could have been prevented if they had been installed.

This accident recalls that at Vogelsaug's Restaurant, Chicago when the welded head of an ammonia generator ruptured at the weld, killing one man and seriously injuring four. Like the Newark generator, explosion occurred before the makers had turned over the apparatus to the purchaser.

The Bureau of Surveys, New York Board of Fire Underwriters, in a report on several ammonia plant explosions gives the following:

By way of accounting for explosions of ammonia vapor suddenly released from high pressure, when not attributable to the presence of lubricating oil, it has been suggested that decomposition of the ammonia might take place, especially in the presence of an intense heat such as that of an electric arc light, resulting in the liberation of hydrogen, which forms an explosive mixture with air. It has also been suggested that certain impurities in the ammonia might increase the tendency to decomposition or explosion. In a number of the cases reported herein a definite explanation of just what caused the explosion of the ammonia vapor seems to be impossible, but the facts as nearly as can be ascertained are submitted.

The practice of testing refrigerating systems with air pressure is especially hazardous, unless all oil is thoroughly removed from the entire system and great care is taken to prevent overheating of the air while being compressed. The last explosion herein reported was due to this cause.

Manhattan Refrigerating Co., Horatio and West St., New York City, accident of Apr. 26, 1915: The refrigerating plant is located on the first floor, and parts of the refrigerating machinery extend to the second and third floors. Parts of the second and third floors were also used as pipe-fitting and repair shops, with an open coal forge on the third. The fourth, fifth and sixth floors were used for cold-storage rooms. The boiler plant was in the basement.

About 1 p.m. a noise described as "a puff," emanating from the

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second floor, was heard by the engineer and other employees. The trouble was caused by the breakage in a pipe connection on the second floor between the analyzer and the rectifier of an absorption-type refrigerating system. Immediately after, a considerable number of glass windows in the front wall of the building were blown out. According to information received, a mild explosion occurred about 5 min. after the first explosion. The ammonia gas had spread rapidly and reached the open coal forge on the third floor, which may have caused this explosion. Apparently the rush of gas scattered the embers of the forge, which resulted in a fire in workmen's coats hanging near-by.

It seems difficult to account for the mild explosion alleged to have occurred about 5 min. after the accident. One theory advanced is that the fire in the forge exploded a mixture of ammonia gas containing various impurities; the explosion can hardly be attributed to oil in the ammonia, as this was an absorption system. It is supposed that the force of the explosion spread the embers from the forge and ignited the combustible material in the vicinity and that this fire caused the two automatic sprinkler heads to operate.

Charles Weisbecker, groceries and provisions market, 267–269 West 124th St., New York City, accident of Aug. 31, 1914: The refrigerating plant consisted of two compressors, one of 40-ton and one of 25-ton capacity, supplying refrigeration by direct ammonia expansion system to the iceboxes in the basement and on the first and third floors. The compressors were chain-driven from 70- and 40-hp. direct-current electric motors. In these compressors oil is injected into the cylinders for the purpose of lubrication and cooling. Shortly after midnight, when the engineer was putting new oil into the system, the oil reservoir with glass sight openings connected to the piping broke and ammonia gas escaped into the room to such an extent as to drive out the engineer. The arrangement was such that practically all the oil in the system would be blown out. There was no evidence of any fire or explosion accompanying the bursting of the glass reservoir. There was no exposed fire or light other than the possibility of a spark at the commutators of the electric motors. No explosion appears to have resulted immediately from the liberation of the ammonia vapor, but there was evidence of fire having occurred later. The place was untenable on account of the ammonia fumes for some time. When entered, the appearance of the compressor room indicated that a mild explosion or burning of the mixture of the ammonia gas and liberated oil had taken place at some time subsequent to the bursting of the gage glass.

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The ammonia safety-valve rules of Massachusetts (effective 1915) are still the subject of considerable comment, favorable and unfavorable. Here again the dollar is the underlying cause. Assume that a builder, particularly a builder of small machines, guarantees the installation for a period long or short. Many such machines as installed in apartments, hotels, etc., use city water tapped off the house supply at any convenient place for condenser purpose. With safety valves provided as required by Massachusetts, it needs only that there be a heavy demand for water throughout the house, and of a duration no unusual, for the condensers to be lacking water long enough to increase the discharge pressure, opening the safety valve and blowing out the charge of ammonia. This happens not infrequently, and with ammonia at 26 cts. per pound, the purchaser immediately demands that the builder make good the charge lost. Certainly this matter is one for adjustment between seller and purchaser and does not concern the law-enforcing body and others, because it is easily preventable mechanically.

Leaky safety valves are another source of disturbance among owners and builders under the Massachusetts rules. And the chief cause of complaint is the loss of ammonia occasioned by the leaking valve. Of all the valves returned to the manufacturer of by far the greater number of safety valves used for this purpose in Massachusetts, red lead, pipe scale, and dirt lodging in the valve, have been the cause of the leakage. Obviously a valve, even though it seats only on a line, will not seat if it comes down on these foreign substances. The legislator would very properly regard this complaint as not entitled to a hearing because the trouble is preventable by reasonable care, which any safety device should receive.

It is probable that the percentage of explosions in ammonia refrigerating plants is small. Statistics, carefully gathered, would show that while there is no need for alarm, on the whole, state or municipal regulations are desirable; further, they would reveal the most common causes of explosions, and in this way lead to quick treatment of the most serious defects. Then too, perhaps the surest and quickest way of gathering

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such statistical matter with the greatest accuracy is through the agency of a law-enforcing body or board, to which all explosions in the jurisdiction are usually required to be reported.

Power has collected considerable information about explosions of ammonia apparatus; it appears that a considerable percentage of such explosions is due to shutting off from the system a vessel containing ammonia, such as a receiver or cooler. The liquid ammonia in such a closed vessel expands, of course, and fractures the container. The art would be benefited if reliable particulars concerning ammonia explosions past and present were to be forwarded either to *Power*, where such accidents could be classified and further investigated, as in this way the profession would have tangable data to guide it in the work of minimizing such explosions.

SYMPTOMS OF TROUBLES IN THE REFRIGERATION PLANT

By Thomas G. Thurston

One of the chief troubles in refrigerating plants is loss of capacity, and the causes are many and varied. In the machine it is generally caused by leaky valves, false heads or pistons. Sometimes the valves become gummed and stick.

Compressor valves should be inspected regularly and the false heads and pistons occasionally. The false head and valves should show a clean seat, bearing all around on a smooth surface. If there are any grooves or signs of the valve or head not bearing at some places, it should be ground or a new one put in. The guides on which the valves move should be a snug fit and be clean and free from gum and grit.

It seems hardly necessary to mention that the cylinder should be smooth, round and free from shoulders at either end. The piston should be a snug fit in the cylinder and the rings should have sufficient tension and be free in the grooves. Leakage at any one of these points decreases capacity.

An insufficient charge of ammonia will result in reduced capacity because it allows gas to get into the expansion coils, and gas in the coils is equivalent to a reduction in active coil surface in proportion to the quantity of gas present.

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When ammonia gas passes from the expansion valve into the expansion coil, from head pressure to suction pressure, the gas is superheated and some of the liquid in the coil must be used to take the superheat. First, the gas escaping into the expansion coil adds to the work to be done by the compressor without producing refrigerating effect; second, some of the liquid available for refrigerating effect is wasted in cooling the superheated gas to the suction temperature.

Oil is a fruitful source of loss in capacity; the oil coats the expansion coils and reduces the heat-transmitting capacity of the metal.

Deterioration of the ammonia will reduce the capacity of the system because it raises the boiling point of the ammonia, and consequently the temperature difference between the ammonia and the cooler or brine will be smaller and the rate of heat transmission slower.

Sometimes the brine gets diluted too much, and when it gets to a low temperature it freezes on the coils and this materially reduces the capacity because the ice acts as an insulator. This will also happen at low temperature, if there is not a good circulation of the brine.

Too much frost on direct-expansion coils will reduce the capacity for the same reason as ice on the coils in a brine tank. The head and back pressures will generally give an indication where to look for trouble. If there is leakage in the machine, the head pressure will drop and the back pressure will rise; if it is due to oil, bad ammonia, insufficient charge or to frost on the coils, the head and back pressures will decrease.

Indication of Insufficient Charge.—Leaky discharge valves can generally be heard if the ear is held close, and if they leak badly the check valve in the discharge line will hammer, the machine will heat and the gage pointers vibrate excessively. A sticking discharge valve will be indicated by the machine racing or running jerkily, the check valve in the discharge line will generally hammer, and the machine will run warmer on that side. Sometimes a valve will stick intermittently, and the machine will have the foregoing symptoms spasmodically.

A false head held off its seat will have the same effect.

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Small leaks can be detected only by inspection of the seat and head; the same is true of a leaky piston. A badly leaking piston will cause the machine to run very hot unless considerable liquid is carried back with the suction gas, whereupon there will be little work done.

A leaky suction valve will be indicated by the frost coming off the leaky side of the machine. If the valve sticks open, the machine will race and the suction-gage pointer will violently vibrate; the gas blowing through can be heard if the ear is held close to the offending valve.

An insufficient charge will be indicated by a sputtering sound in the expansion valves, and the frost line on the expansion valves will disappear; and if the valves are opened too much in an attempt to get the frost back to the machine, the valves, liquid header and the liquid line for some distance back will become frosted. This is because the expansion valves are opened so wide that the liquid line is unable to furnish the supply. The head pressure and back pressure will come down the frost will disappear from the suction side of the machine, and the latter will run hot. Oil, poor ammonia and frosted coils will all have the same effect. The back and head pressure will drop, and the expansion valves will have to be closed more to accomodate the slower rate of evaporation.

Excessive head pressure is another frequent trouble. This is caused by air or foul gas in the system, condenser badly scaled on the water surface, oil in the condenser or some of the condenser coils blocked. In a double-pipe condenser the water pipes are readily scaled if the water is not of the best; the annular space between the pipes is small, and this sometimes blocks with oil, scale and other foreign matter. Generally, if this happens a number of the stands will be cold, and by going along the condenser and feeling them, the "dead" ones can be located. These should be shut off with the water running on them and left standing for a few hours and then purged. This will decrease the head pressure. If the condenser is too small or the water warm or insufficient, high head pressure will be practically impossible to avoid.

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BACKING OFF THE GLAND

By A. C. McHugh

Various methods of determining the distance the gland on the piston-rod stuffing box of an ammonia compressor may be safely backed off are in use. I believe a satisfactory method is to file a flat spot on one or two of the threads at a point where they will correspond with the face of the gland when it has been backed off as far as safety permits. It is then easy to determine the relative position of the gland even in the dark, by simply placing the fingers on the threads. Of course the value of this or any other means exists only in the observance of it, as the following incident will show:

Before I took over the watch, one of the engineers showed me about the plant to familiarize me with my duties, and in speaking of the compressor, which was not running at the time, he told me if the rod should run warm I might back off the gland. I asked how far it could be backed off safely, and he said there were two flattened threads which, when uncovered were supposed to indicate the extreme point of safety, but that he had backed the gland off beyond them several times without trouble. To demonstrate he began to back the gland off, leaning over it to observe its position, when suddenly the gland blew off, the packing and oil lantern were forced out and his face and hands were covered with ammonia-saturated oil, which blinded him for sometime.

We replaced the packing and oil lantern and screwed the gland on again, thinking no damage had resulted; but when I started the machine, I noticed that the oil level in the reservoir of the circulating system feeding the piston-rod gland was dropping rapidly. I took up more on the gland, which resulted in the rod running warm, but the oil level still continued to drop. Being unfamiliar with the machine, I was puzzled but managed to get through the watch by refilling the reservoir several times. Before going home, I reported the loss of the oil, but said nothing of the accident. The machine was repacked during the day, but upon starting again the next night I began to experience the same trouble. This

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time I noticed that a squirt of oil accompanied each outward stroke of the piston rod, and the oil level dropped so fast that I saw it would be necessary for me to do nothing else but pump oil into the system if I was to hold the level. I called up the chief, who came over and took a look for himself. We then shut down the machine and examined the rod, in which we found a nick, the back of which formed a knife-edge protruding 1/16 in. from the side of the rod. This edge working back and forth had cut a groove in the packing. I knew immediately that the nick had been formed when the gland had blown off.

A GAS ENGINE DRIVEN COMPRESSOR ACTS UP

By D. L. Fagnan

After chasing Sam Bernard around the country for half a season, we were both in the same town on the same day and I had phoned and secured B-4 orchestra seats for that evening. The razor was keen and I gleefully took off the second coat when—. Well, 'twas a wire from the office. The Jones Cold Storage Co., Blankton, Tex., 800 miles away, was in trouble. Leave at once. Fifty minutes later I was aboard the Limited and again Fate, steam and steel were increasing the distance between Sam and me.

The Jones plant was typical of those in Texas. A 50-ton compressor was driven by producer-gas engines. One 100-hp. 3-cylinder and an 80-hp. 2-cylinder engine were belted through friction clutches to the mainshafting. The layout was faulty in not having a friction clutch placed on the mainshaft to enable starting with a free shaft, gradually cutting in the compressor.

The trouble had to be quickly overcome, as some days it would be 9 or 10 a.m. before the plant could be made to run. Sometimes the engines would get a few good explosions and then pound. There would be heavy thumping in the compressor cylinder due to liquid ammonia. Then the operator would try to check the speed of the gas engine to minimize the thumping in the compressor.

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I spent a day at the plant looking things over and laying out a plan of action. First we wired for the necessary fittings to make the bypass pipe full size. These arrived the following day by express and were installed that night, tested out and found all right. A friction clutch of the 6-arm type, known not to "lay down" on any job, was ordered. Of course it was ordered split. This was next installed, and as we had a fairly slack belt, we were able to easily run it off and on the pulleys.

Next the air cams on the engine were checked to see if full pressure of air was admitted at the proper time, namely, about 2 to 3 deg. early. We found them admitting 9 deg. early. The early admission when operating at such slow speed at starting neutralized a great deal of the energy of the air pressure. The camshaft was removed, and new offset keys were made and driven after careful measurements with a protractor had demonstrated that the angle was correct within ½ to 1 deg. The other engine was found to be taking air 12 deg. late, and as the exhaust valve opened at 44 deg. from the lower center, one can imagine what waste of air had occurred when I say that from a starting pressure of 160 lb. the pressure after one revolution was 100 lb. and after the second or third, about 50 lb., which was not sufficient to cause the engine to go over the top center. We reset the cams to take in air at 3 deg. early, which corrected the trouble had with starting.

Ignition System Needed Adjustment.—Next the ignition systems were thoroughly gone over, one at a time. We found ignition taking place 18 deg. early at starting and when operating, 34 deg. early. I changed this to 7 deg. early for starting and 24 deg. for operating, with provision for easily varying the degrees by protractor readings on the shaft, leaving permanent marks; also on the eccentric cam I stamped degree marks.

The other engine ignited 10 deg. late at starting and about 12 deg. early when operating. This was changed to 5 deg. early at starting and 22 deg. early when running, which caused the engines to pull the load so much easier that we found we had ample power using either engine.

The next problem to solve was the reason why the operator could not get the proper temperature in the calcium-chloride

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brine tank. He could seldom get better than zero deg. F. He informed us that he carried 15 lb. back pressure. There were two 25-ton vertical brine coolers. I determined to carry a low back pressure—about 2 lb.—and this gave a brine temperature of 15 deg. F. The frost was allowed to nearly cover the top pipe leading to the gages, showing the cooler contained liquid ammonia up to about two-thirds full.

One may ask why they had to carry such low back pressures to get a low brine temperature. The engineer at the plant thought that the more liquid ammonia allowed to go to the brine coolers the lower the brine temperatures would be. He said he was told so by the erecting man who had installed the plant. It is true that 15 lb. back pressure corresponded to about zero deg. F., but to get the proper flow of heat from the brine to the ammonia it was necessary to carry an ammonia temperature lower than zero, say from 5 to 6 deg., preferably 8 to 10 deg., so as to get the required temperature head.

AMMONIA LODGED IN CONDENSER

By V. O. Safranck

Recently, I took charge of a plant that has been noted for its mysterious disappearances of ammonia, and I have had considerable trouble keeping enough ammonia in the receiver, notwithstanding that the plant had been recently overhauled by an expert.

After charging 350 lb. more ammonia into the 1,500-lb. receiver and not finding a trace of it, I began investigating and finally decided that it was in the condensers even though I could not pump it down from there. I put oily waste in the pan under the condenser coils and warmed them with a light blaze. The liquid soon filled the receiver

But the next day the same thing happened again, so I decided that the fault was with the design of the condensers. I pumped down two coils out of five under a light temperature to facilitate the movement of the liquid from them. After thoroughly pumping down, I cut them out and then ran on

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three coils. Since then the receiver is almost full while running and the head pressure is lower. Before cutting out the two coils this pressure was from 120 lb. to 180 lb.; now it sometimes comes down to 105 lb.

COMPRESSOR VALVE STUCK OPEN

The machine in question consisted of two vertical, single-acting compressors driven by a simple non-condensing Corliss engine. One day the machine began to race badly, but the engineer could find nothing wrong with the governor and valve gear, and he finally had to throttle the machine, but this did not completely stop the racing. About this time he noticed that the crankpin of one of the compressors was smoking. He stopped the machine, let the pin cool and forgot about the first trouble. When he started again the machine ran all right, but a few days later the same thing happened, and this time he also failed to notice anything wrong until he saw the smoking pin.

It was not very many days before the same thing happened again, and this time he throttled the engine at once and directed his attention to the crankpin, which became hot. This happened several times, until, looking at the ammonia gages one day, he noticed that they were vibrating much more than usual. Paying closer attention, he noticed that they worked in the reverse direction at the same moment; that is, the head pressure would increase and the back pressure decrease at the same time, and then the reverse would occur. There was only one vibration for each revolution of the machine.

After thinking this over, he concluded that only one compressor could be working, because if both had been working there should have been one vibration for the discharge of each compressor, or two for each revolution. Going on top of the machine, he found the frost thawing rapidly on the suction line to the compressor that had been giving trouble. Also there was a whistling sound in the discharge valve of the compressor and the cylinder and piston rod were much hotter than those of the other machine.

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On removing the valve, the cause of the trouble was evident —the valve was stuck open. Thus full head pressure was maintained on the piston during the whole revolution, and this put extra pressure on the crankpin, causing it to heat. This also explained the racing of the machine, as it was working only on one side at times, as the valve obviously stuck intermittently. The valve also evidently stuck when it got hot, and this would explain why the machine would run all right after it had been shut down for a while. The valve was dry and badly gummed, but when cleaned and oiled the compressor gave no further trouble.

ON THE JOB WITH THE REFRIGERATING TROUBLEMAN

By E. W. Miller

One day the engineer of a small packing-house plant, who by the way had been on the job only a few days, was surprised to find that the back pressure on one of the ammonia compressors had dropped from 25 to about 10 lb. Another smaller machine running on the low-temperature rooms went along well. At the same time the machine on which the pressure had dropped began to freeze, and the temperatures in the coolers began to increase.

I dropped in to visit the engineer a few days after, and he related his troubles. Becoming interested, I offered my services, which were gladly accepted. We made a trip around the plant, and the one thing that struck my attention was the small opening given the expansion valves, even those on some very long coils. All the rooms carried by this machine were above 35 deg., and accordingly the expansion valves on some of the coils should have been open at least half a turn, as the valves were only ½ in. and some of the coils were over 3,000 ft. long.

The engineer informed me that he had to "pinch back" on the expansion valves from the day that the back pressure had gone down. This sounded unreasonable. Normally if the back pressure goes down and the room temperatures remain the same or go up, as they did in this case, the expan-

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sion valves can be opened a little more; for when the pressure goes down, the ammonia drops in temperature, which increases the temperature difference between the boiling ammonia and the room, and consequently the coils will evaporate more ammonia. Also, if the back pressure goes down with the room temperatures remaining the same, the machine always warms up until the expansion valves are opened more. In this case the machine froze, while the cooler temperatures went up.

Pressure High in Evaporating Coils.—Now it was plain from the amount the expansion valves were opened that the coils were not evaporating the quantity of ammonia they should with the suction pressure carried. As the coils were comparatively clean, there could only be two reasons for this: The coils might be loaded with oil and so be incapable of doing their work; in this case, however, the change would have been gradual instead of happening suddenly, as it did. Or, what looked to me like the most reasonable solution, the pressure in the coils was too high, in, which case the coils would not evaporate sufficient ammonia as the temperature difference between the rooms and the ammonia would be decreased. By the same reasoning, if the pressure in the coils was excessive, there must be some place in the suction line where the gas was throttled to bring it down to the low suction pressure.

We went back to the engine room, removed the suction gage and connected it to one of the expansion coils. To our surprise it registered over 30 lb. As all the coils were acting the same, I concluded that the trouble must be somewhere in the suction line. We started back along the suction line, thinking that we might hear where the throttling was taking place. When we got back as far as the engine room, we could hear a hissing sound, like steam passing through a partly closed valve. Further investigation located the sound in a stop valve in the suction line some distance from the compressor.

We immediately pumped down the system and opened up the valve. The solution of the trouble was quite simple. The valve was of the angle type and had been placed in the line with the stem horizontal, in which position, in this line, the flow was against the disk. The disk had rattled loose from

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the stem and had gradually worked up to the seat, thus throttling the gas. I advised the engineer to reverse the valve at the first opportunity.

A 250-ton vertical single-acting twin-cylinder compressor had been thoroughly overhauled. After it had been put back into service for a few days, it developed a slamming knock, first in one cylinder and then in both, though when the cylinders were opened and everything checked they were found to be tight and solid.

When the machine was started again, it knocked as much as ever. Thinking the knock might be caused by too little clearance, which had been adjusted to $\frac{1}{32}$ in., the clearance was increased, first to $\frac{1}{16}$ and then to $\frac{1}{8}$ in. without results. The suction valves are located in the piston, and their weight is balanced with a spring that tends to open the valve, which is practically opened by its own inertia as the piston starts on the return stroke and requires little pressure to open it. Thinking that this valve opened so freely as to come up far enough to strike the head, we removed the spring from the valve, but without the desired results.

Springs too Weak in False Head.—It was peculiar that the knock did not seem to occur just when the crank passed the center, but after it had moved some distance past the center. The machine was equipped with false heads; and when these were being replaced, the oiler suggested that the springs holding down the false heads might be too weak. At this suggestion, enough washers $\frac{1}{2}$ in. thick were made to go under all the springs of one head. When the machine was brought up to full speed, the side that had been supplied with the washers ran noiselessly, while the other knocked as badly as ever. When washers had been put under the springs of this side in the same manner, the knock disappeared.

Another Experience.—Responding to a hurry call from an engineer friend, I found that the chief trouble was that the condenser pressure was unusually high, about 225 lb. The relief valve was set at 235 lb. I suspected that air and foul gas had accumulated in the condenser, and I asked the engineer if he had purged the air out of the condenser lately. He

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admitted that he had not done this since the plant was started. We spent nearly all the afternoon getting the air out of the condenser, but the pressure had dropped to only 205 lb. I was unable to account for this, as I was certain that all the air was out and that every coil was working, as was evidenced by the even temperature of the whole condenser. I began to suspect that the condenser pressure gage was not telling the truth. To satisfy myself, I removed it and then opened the valve in the gage line a little to see if there was any pressure on the line. As I had suspected, there was a strong vacuum on it. The gage, after it was removed, remained at 60 lb. The condenser pressure, instead of being 230 lb., had been about 170, which was not so bad. I later determined that there was a variation of 5 lb. from zero to 150 lb. between the two gages, but this was close enough for practical purposes.

When I removed the suction gage it remained at 20 lb. To test it, I connected a tee to a connection on the condenser pump. On the branch of the tee I connected a 100-ft. length of garden hose, and on the other end of the run I connected the gage. The hand on the gage was then set back to zero. I then figured out the height of a column of water that would give a pressure of approximately 25 lb. and elevated the end of the garden hose to this height. Then the water was slowly turned into the hose until it escaped from the top. The gage showed about 1 lb. higher than I had figured, but this was close enough.

As soon as we got started, I went around with the engineer and set the expansion valves about where I thought they ought to be. When we got back to the engine room, we found the suction pressure had gone up to 18 lb. As none of the rooms were carried below 10 deg., this was just about right for rooms that were similarly piped. Next, we went after the condenser and purged the air out of it again. When we got through, the condenser pressure had gone down to 135 lb. An explanation of the trouble was now quite simple. The excessive condenser pressure in part explains itself. Due to the defective suction gage and the engineer's trying to main-

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tain a certain suction pressure, the system had been running on a vacuum and had run the stuffing boxes too loose. The machine was of the single-acting type, so there was no pressure on the packing. Consequently, the air had leaked in around the packing, and this had run the condenser pressure up as it had accumulated in the condenser. To maintain the low suction pressure, the engineer was obliged to nearly shut off the greater part of the coils, and this was the reason he was unable to keep the temperatures down.

OIL TROUBLES—HOW TO AVOID THEM

An oil separator in the compressor discharge line, and too close to the compressor will not work properly, as a rule, as much of the oil at this point is vaporized. Locate the separator near the condenser.

Put a cock close to the separator on the oil blowoff line in addition to the valve on this line. If a bit of packing should jam in the latter valve the flow may be stopped by closing the cock.

Unless the bushing in the back end of the piston-rod stuffing box of the compressor fits closely around, the rod packing will be drawn into the system together with oil.

See that the oil lantern fits snugly in the stuffing box, otherwise it will bear heavily on the piston rod, wearing it.

Connect an oil trap to the lowest point of direct expansion piping, and to the lower manifold at brine and ice tanks.

To clean piping and coils of oil, blow steam through them to soften the oil and flow it out. Then blow air through to dry the piping. When about to blow piping or coils open a flange at the lowest point, if a flange is there. If the coil is continuous, remove it and place so the oil and condensate will drain out.

Clean piping and coils of oil at least every 3 years, and better every winter when the load is light.

HOW TO READ THE HYDROMETER

The correct method of reading the hydrometer is illustrated in Figs. 82 and 83, reproduced with the directions

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for reading from Circular No. 57 of the United States Bureau of Standards. The sample of oil is placed in a clear glass jar or cylinder and the hydrometer carefully immersed in it to a point slightly below that to which it naturally sinks, and is then allowed to float freely.

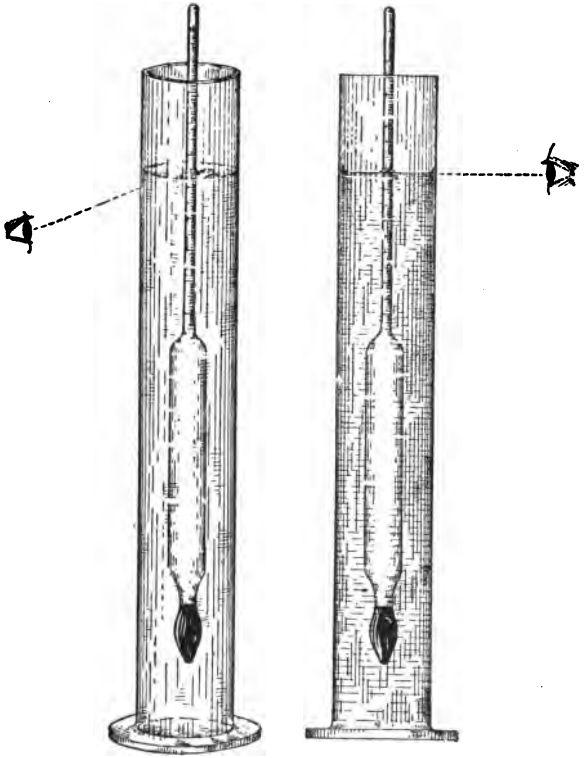


FIG. 82. FIG. 83.

FIGS. 82 and 83.—How to read hydrometer.

The reading should not be taken until the oil and the hydrometer are free from air bubbles and are at rest. In taking the reading the eye should be placed slightly below the plane of the surface of the oil (Fig. 82) and then raised slowly until this surface, seen as an ellipse, becomes a straight

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line (Fig. 83). The point at which this line cuts the hydrometer scale should be taken as the reading of the instrument (Fig. 83).

If the oil is not sufficiently clear to allow the reading to be made as described, it will be necessary to read from above the oil surface and to estimate as accurately as possible the point to which the oil rises on the hydrometer stem. It should be remembered, however, that the instrument is calibrated to give correct indications when read at the principal surface of the liquid. It will be necessary, therefore, to correct the reading at the upper meniscus by an amount equal to the height to which the oil creeps up on the stem of the hydrometer. The amount of this correction may be determined with sufficient accuracy for most purposes by taking a few readings on the upper and the lower meniscus in a clear oil and noting the differences.

A specific-gravity hydrometer will read too low and a Baumé hydrometer too high when read at the upper edge of the meniscus. The correction for meniscus height should therefore be added to a specific-gravity reading and subtracted from a Baumé reading.

The magnitude of the correction will obviously depend upon the length and value of the subdivisions of the hydrometer scale and must be determined in each case for the particular hydrometer in question.

MAKING A GLAND NUT SAFE

An ammonia compressor was repacked, and the rods ran quite warm for a few days. The screw gland was backed off so far that it made one man nervous for fear it would come off if he loosened the gland to make the rod run cooler.

It was proposed to drill three holes equally spaced around the nut and down from the edge far enough so that if iron was felt through the hole the gland was on far enough to be safe, but if not it was unsafe to back off any more. The gland is shown in Fig. 84.

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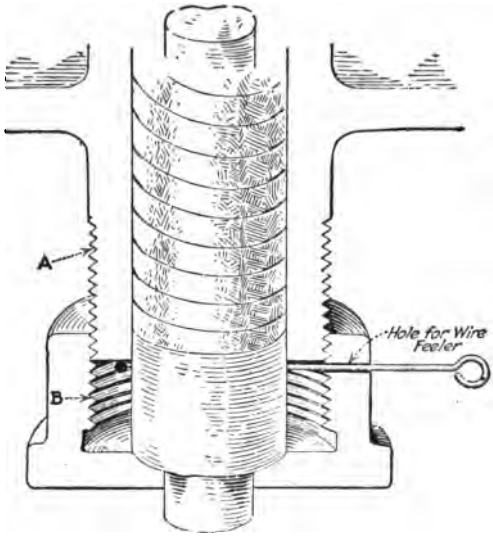


FIG. 84.—Rod gland.

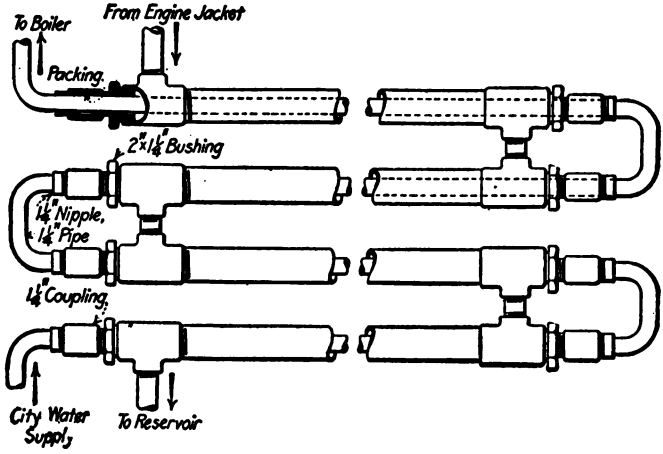


FIG. 85.—Recovering waste heat.

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JACKET OVERFLOW HEATER

In an oil-engine-driven refrigerating plant it was found desirable to utilize the waste heat of the jacket water. This might have been done by installing a small pressure pump to handle the water from the jacket overflow. However, as the purity of this water was somewhat questionable, the engineer devised means by which this heat could be transferred to water already under pressure from the city mains.

Four 18-ft. lengths of 2-in. pipe were made up as shown in Fig. 85. Inside of these, four 20-ft. lengths of 1-in. pipe were inserted, with the ends projecting beyond the tees at the ends of the 2-in. pipe. Packing boxes were arranged by bushing the open ends of the 2-in. tees to 1¼ in. Then with a 1¼-in close nipple, a pipe coupling and a shorter piece of 1¼-in. pipe threaded at one end, the packing box was complete (except for packing).

FORCING OBSTRUCTIONS FROM AMMONIA TRAP LINES

The sketch in Fig. 86 shows a pump made for opening the plugged blowoff pipes to amonia traps when they became stopped with scale or other matter. One day when I attempted to blow out the oil on the high side of the machine, I found that it was stopped so tightly with scale and rust that the pipe was tight against the air pressure of 180 lb. with the valve wide open. I decided to force the scale and rust back into the trap and for this purpose made the pump shown. The ammonia lubricating oil was the liquid pumped into the line against the obstruction. Note that the oil is to be poured into the suction side of the pump, which may be so adjusted as to stand vertical.

REMOVING SEDIMENT FROM TANK

An easy way to clean dirt out of a tank when there is water overflowing is shown in Fig. 87. Usually the overflow pipe is connected to the side of the tank at the top and it skims the water, but the sediment that settles on the tank's bottom is not removed. By extending the pipe inside of the tank to the

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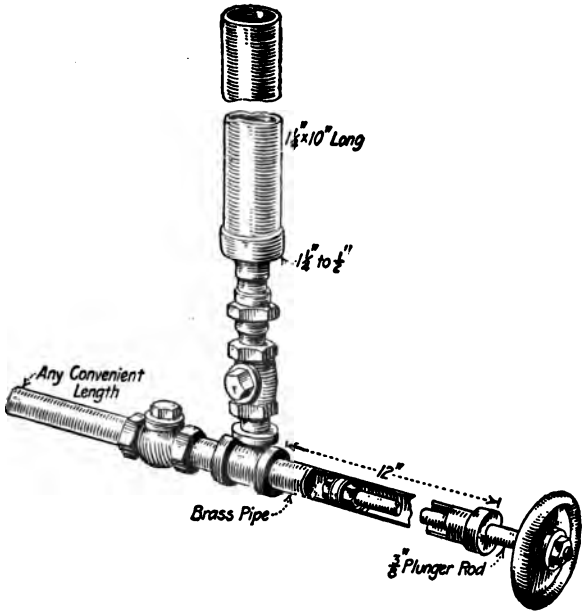


FIG. 86.—Force pump for cleaning piping.

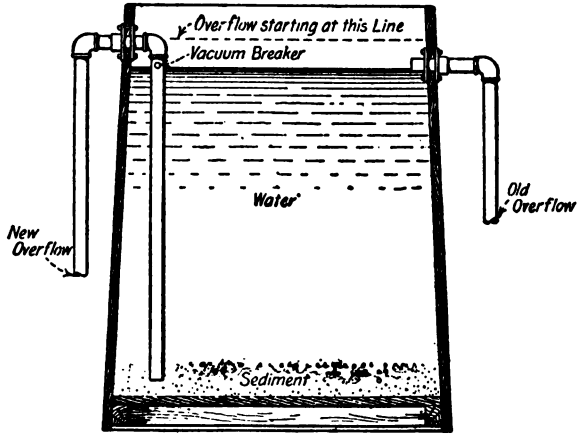


FIG. 87.—Cleaning sediment from tank.

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bottom, as shown, and drilling a hole in it at the proper level, any desired amount of water can be discharged and it will carry a large quantity of dirt from the bottom of the tank. The hole should be located at the lowest water line that it is wished to carry otherwise the tank will be emptied, because the pipe forms a siphon.

OILING AMMONIA-COMPRESSOR ROD

The illustration in Fig. 88 shows the way an engineer has kept the piston rod oiled on an ammonia compressor for the last 3 years. The oil reservoir is set to one side of the lantern, with the bottom of the sight glass level with the bottom of

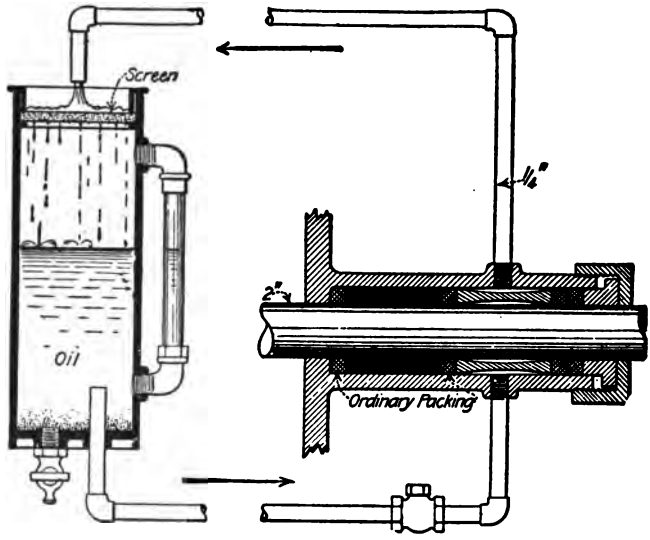


FIG. 88.—Oiling compressor rod.

the rod. The oil will then circulate from the bottom connection of the reservoir, through the lantern and back to the reservoir by way of the upper or discharge pipe. This is a whole lot better and less trouble than the old-style pump run from the shaft by a belt.

The oil is circulated by the action of the rod going back and forth through the oil. On the out stroke oil is drawn through

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the bottom connection from the reservoir, and on the other stroke the oil goes out at the top pipe and back to the receiver.

THERMOMETERS IN THE REFRIGERATING PLANT

They are essential to intelligent, efficient operation. Many experienced engineers prefer chemical thermometers inserted in wells in the water, brine or ammonia pipes.

For a distilled-water ice plant there should be five thermometers on the inlet and outlet of the cooling water lines, on the inlet and outlet of the distilled water, and on the outlet of the storage tank.

Place a thermometer in the liquid line leaving the ammonia condenser and in the water line to the condenser; put one on the waste line too. A therometer should be inserted in the liquid line at the expansion valve. If the ammonia receiver or the liquid lines are exposed to heat, insulate them.

Always have a thermometer in the suction and discharge lines of the compressor.

UNUSUAL CAUSES OF RACING OF AMMONIA COMPRESSORS

By E. W. Miller

Recently one of the ammonia compressors in our plant began to give considerable trouble because of racing. The machine was a 100-ton having two vertical double-acting compressors with the engine vertical and set between the two compressors. Sometimes it would run along for a number of days without trouble, and then it would suddenly start off and race until the safety cams operated and shut down the engine unless someone happened to be near and managed to close the throttle. This happened a number of times; the governor, the whole machine in fact, was examined without finding the cause of the trouble.

Being somewhat at leisure one Sunday, I decided I would stick close to the outlaw during the day, particularly as it had shown indications in the morning of starting its regular capers. For the greater part of the day nothing happened. I was oiling the valve gear of the engine on the deck of the machine

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when trouble started. First the machine would speed up and then suddenly slow down until the valves on the engine failed to unhook. Then it would speed up again at a greater rate before it would swing to the other extreme and slow down. The throttle was closed enough to hold the speed down to a safe limit. I then had one of the assistants watch the throttle while I investigated the governor. This was in perfect condition; even the plates on the hooks and arms of the valve gear were in excellent order. These in many cases had caused similar troubles before, but not so in this case.

Sticky Valve Stems Cause Trouble.—While I was examining the engine valve gear, an unusual slam in one of the compressors attracted attention. As near as could be judged it was a valve slamming down on its seat. At the same time the machine nearly stopped. With this as a clue I found a convenient box and settled down for a still watch for trouble. The wait was not long. The moment the machine started its capers gas could be heard whistling through one of the valves, and by a few smart raps on the cage the trouble was corrected before the man watching the engine could get to the throttle. Evidently, a valve was sticking.

When we removed the bonnets of the top valves of the compressor, it was found that the valve stems were dry and covered with a rather hard, sticky substance; this evidently had caused the valves on the top of the cylinders to stick at times and possibly on both compressors at the same time in some cases. It is presumable that when one of the valves stuck and unbalanced one of the compressors and the machine began to speed up, this intermittent action of the valves caused them to open more violently and move farther than normally, causing both to stick for short periods. If one or both valves let go at the same time and resumed normal operation while the machine was running at high speed and the governor at the top of its travel, it would slow the machine down so much that the safety cams would act and shut it down altogether.

The sticky substance was made up of oil and dirt, and when the discharge gas was very hot they baked on the

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valves and stems. It was apparent from the appearance of the stems that little oil reached the valves, as they were practically dry and had a high polish in places. There was reason for this as the oil supplied to the machine was applied to the stuffing-box lantern and of course all the oil that got into the compressor was that carried around the system and that which was carried into the cylinders with the piston rods. The oil that was drawn in by the piston rods went only to the bottom valves and that drawn in from the system went the same way, as it would naturally drag along the bottom of the pipes and go to the lower end of the machine where the branches to the suction valves were made.

To remedy the trouble we first cleaned the valves and stems with gasoline and put them back in place. We then found two pressure oil cups that formerly had been used on the lanterns of the machine and drilled holes in the valve bonnets and connected the oil cups into them. By regulating the feed to about two drops per minute, the valves got plenty of oil and no further trouble was experienced.

Valve Disk in Suction Line Worked Loose.—In another case one of the machines began to run away without any apparent reason. Everything would be going along nicely when, without warning, the machine would speed up at a rate that made it necessary to shut it down. This happened two or three times; the most thorough investigation failed to show anything wrong.

One day one of the engineers happened to be looking at the suction gage at the moment the machine went speeding. To his surprise the pressure went down to about 20 in. vacuum in a few moments. Here, evidently, was a clue to the trouble. Some place or somehow the line was shut off, and of course with no gas to pump there was no load on the machine, and it naturally got frisky. In the meantime one of the oilers had stopped the machine and started it again. The suction pressure had returned to normal a few moments after the machine stopped, and when the machine was again started it remained at normal, thus indicating that whatever the trouble was, it had disappeared when the machine was stopped.

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Some of the crew wanted to pump down and investigate the machine from stem to stern, but one of the assistant engineers insisted that the trouble could not possibly be in the machine because machine trouble would not cause the suction pressure to drop in the manner described. The force then directed their attention to the suction line. About 25 ft. back from the machine was a stop valve in the suction line, placed in horizontal position. Some one suggested that this valve might possibly stand investigation. So the line was pumped down and the bonnet of the valve removed. A glance was sufficient to disclose the cause of the trouble. The valve had been placed in the wrong position, and the flow through it had a tendency to close the valve against the seat. The disk had worked loose from the stem, and as the valve was of the globe type and lying on its side with the stem extending out in a horizontal position, the disk had been dropped to such a position that with a sudden rush of gas through the valve it would swing against the seat and the pressure would hold it in place. As soon as the machine was shut down there was sufficient leakage past the valve to fill the line again, when the disk would drop back to its regular position. To prevent further possibility of trouble from this source, the valve was reversed so as to bring the flow of the gases against the bottom of the disk, which would tend to open instead of close the passage in case the disk should be dropped at some other time.

Compressor Raced When Springs Broke.—In another case a 150-ton vertical compressor developed a serious case of racing. It would run along well for a few minutes and then start to cut up capers for a few revolutions. As usual, we did everything possible to the governor and valve gear of the engine but without results. The next move was to look into the valves of the compressors. We found nothing wrong in the first compressor, but got no further than the removal of the cylinder head of the second before we located the trouble. Four of the heavy springs holding the false head in place had broken, allowing the head to lift from the seat. At this time it had evidently got caught while partly cocked to one

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side, with the result that it had failed to seat at all, and of course the gas rushed back into the cylinder as fast as the piston went down. This is most likely what happened before when the machine began to race. Anyway, after putting in new springs in place of the old ones we had no further trouble.

TESTING PLANT WITH AIR

Care is necessary to avoid explosion. The heated lubricating oil in the system is a hydrocarbon, and when properly mixed with air forms a powerful explosive, particularly because of the hydrogen in the ammonia. Due to compression when testing the system may become dangerously hot.

A thin coating of lard oil should first be applied by hand to the walls of the compressor cylinder and the compressor run until the pressure reaches 100 lb. Now shut down long enough to cool down, then start compressing until about 50 lb. additional pressure is had. Stop again and cool down, compressing slowly so as never to get the discharge line so hot the hand cannot be held on it.

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CHAPTER VIII

THE ABSORPTION REFRIGERATING SYSTEM

There is a saying among salesmen for refrigerating machinse that it is easier to teach a man without any idea of engineering about the care and operation of the absorption refrigerating machine than it is to get a licensed engineer to learn these things. This statement is far-fetched, and yet there is something in the remark. Perhaps it is that the absorption machine is a mass of valves and fittings; perhaps it is that there are no moving parts except a small pump hidden off there in the corner, or perhaps the engineer is hunting for something hard and overlooks the fact of the simplicity of this form of refrigerating machine. Be this as it may, the writer has figured the matter out for himself, and the following is his own explanation, offered to the reader in the hope that it may be of some assistance in understanding and operating this important machine.

The absorption machine is different from the compression machine and yet it obtains the same results. It has to increase the pressure to the same amount if the ammonia from the expansion coils is to be condensed. And yet it is different because it has no pump to compress the gas. In fact, the absorption machine uses only heat to cause the increase in pressure—heat from boiler steam or exhaust steam or heat from any other source. The compressor is replaced by a distilling apparatus, more like the stills used in fractional distillation of alcohol or lighter grades of oil. This ammonia still contains a solution of ammonia and water. It must be understood that water can dissolve a certain amount of ammonia, depending on its temperature and the pressure exerted on it, and that it may be concentrated with a certain percentage of ammonia when the pressure and temperature are of a certain

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amount. Increasing the pressure or decreasing the temperature will enable the water to absorb more ammonia. Likewise, if a solution has its temperature raised and the pressure is kept the same, there will be a point where some ammonia will be driven off as a gas.

Action of the Generator.—The action of the still, or the so-called generator, is shown in Fig. 89*A*, which represents a solution of ammonia and water (called aqua ammonia) con-

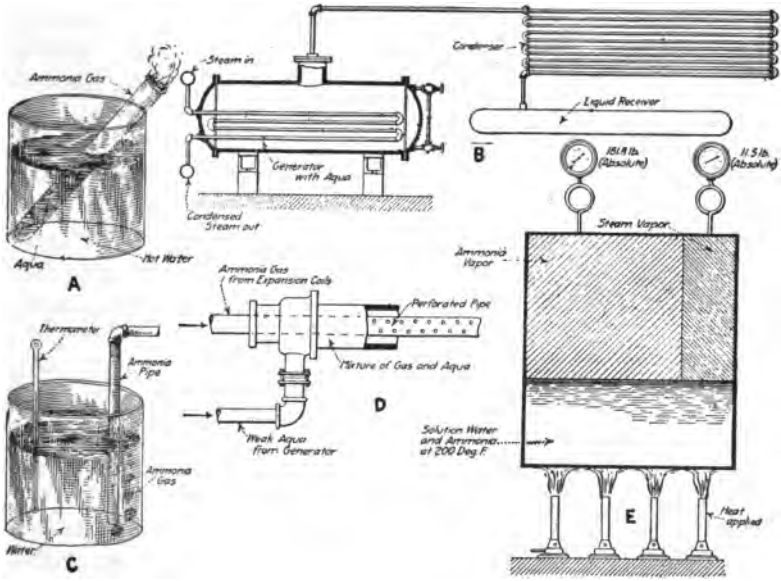


FIG. 89.—The elementary absorption machine.

tained in a glass tube, and all immersed in a glass of hot water. The heat taken up by the aqua drives off some ammonia gas as shown, and the aqua which remains is one of less concentration. In Fig. 89*B* the absorption machine is more nearly represented. The large vessel contains the aqua, which is under a pressure determined by the action of the condensers. The coils within this vessel are the steam-heating coils which keep the aqua hot and provide the necessary heat for driving off the gas into the condenser, in the same manner as in Fig. 89*A*.

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As the action continues, the concentration of the aqua would become less and less and finally the ammonia gas would be entirely removed unless means could be found to keep the concentration up to full strength. But first let us see what takes place after boiling the liquid ammonia in the expansion coils.

System May Operate Direct Expansion or With Use of Brine.—The gas enters the condenser as a slightly superheated vapor under pressure (similar to the action of the compression cycle) and is condensed. The condenser, receiver and expansion coils are substantially identical with those for the compression machine because the liquid ammonia is likewise identical. But there is no compressor to take the low-pressure gas from the expansion coils. The absorption refrigerating system may operate direct expansion or with the use of brine. In the latter case the expansion piping is arranged for in a so-called brine cooler. The condenser cannot condense it again into liquid ammonia unless its pressure is increased some 6 to 12 times. It is necessary to get this gas into the generator again. It could be pumped in as *gas*, but if that were done, we might just throw away the generator shown in Fig. 89*B* and use a compressor. Therefore, it is clear that some convenient and economical means must be obtained for getting the ammonia gas into the generator. This is done by dissolving the gas in water and forming a so-called strong aqua.

Water has a great ability for absorbing ammonia. It will dissolve several hundred times its volume of the gas. For instance, 1 lb. of NH_3 (ammonia) at standard pressure, 15.25-lb gage, will occupy 9.2 cu. ft. If weak aqua of 20 per cent concentration is brought over from the generator (that is the only water available in the absorption system) and this pound of gas is allowed to dissolve in it, say at 10-lb. gage and 90 deg. F., there would be required about 6.9 lb. of the resulting mixture to be pumped into the generator. This would occupy, roughly, 0.115 cu. ft., or about one-eightieth as much as if the gas as a gas had been compressed in a compressor under standard suction pressure. And it is evident that the size of the pump (the piston displacement required) is not affected by

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change of boiling temperature of the ammonia in the cooling coils. The usefulness of the absorption machine for low-temperature work is therefore self-evident.

The foregoing action can be shown by Fig. 89*C* where gas is allowed to bubble into a part of water. The gas is quickly dissolved so that the bubbles do not reach the top. In the absorption machine the system is a closed one, and therefore it appears more like Fig. 92, which shows the usual type of atmospheric absorber. Here the inner pipe extends part way the length of the stand and is perforated as shown in Fig. 89*D*. Weak aqua enters on the bottom left side and mixes with the gas, and the current carries the mixture and dissolved gas into the upper pipes. But whereas in *A* and *B*, it required heat to cause the gas bubbles to occur, yet the reverse action takes place in Figs. 89*C* and *D* and heat is generated when the dissolving action which forms strong aqua takes place, and the thermometer shown in Fig. 89*C* will indicate the rise in temperature as heat is evolved during solution. However, a rise of temperature could not be allowed in Fig. 90 or a temperature soon would be obtained which would prevent further dissolving of gas. So this heat

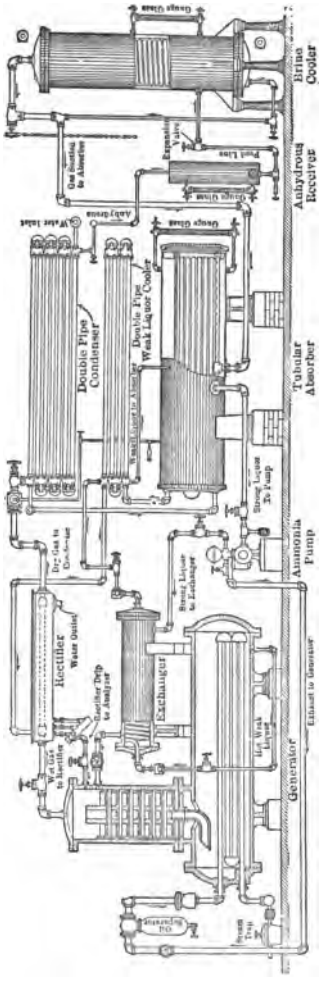


FIG. 90.—General arrangement of tubular type absorption refrigerating machine.

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of solution is removed as promptly as it occurs by means of cooling water, shown at the right in *B*.

The strong aqua pump then takes the strong aqua from the absorber at a pressure slightly higher than the boiling pressure of the ammonia in the expansion coils, and pumps it into the top of the generator, where it mixes with the aqua there. As weak aqua is heavier than strong, the solution at the bottom should be the weakest. It is customary to connect the bottom of the generator with the line going to the absorber, in order that the necessary liquid be obtained to dissolve the gas. The elementary absorption system, then, consists of a generator where the ammonia gas is driven off under condenser pressure, a condenser where the gas is condensed, a set of expansion coils where the gas is vaporized, an absorber where the gas enters a solution again, and finally an aqua pump to drive the strong aqua into the generator. The machine could work under these conditions, but not very economically.

One additional piece of apparatus required is the so-called exchanger, an apparatus required for reasons of economy. The strong aqua is cold—as cold as the water used in the absorber will allow it to be. To put cold strong aqua directly into the generator would be an unwise as to pump cold feed water into a boiler. Likewise, the weak aqua leaves the generator at a temperature somewhat less than the temperature of the steam used in the steam coils. It would be impossible to get satisfactory action in the absorber until the temperature of this aqua is reduced, say, to 100 deg. F. or less. Therefore, by the use of an interchanger, designed for counterflow, the strong aqua takes heat from the weak aqua, and is made hot while the weak aqua is cooled.

Let us consider Fig. 89*E*, which shows what really takes place. As the aqua is heated by the steam coils, its temperature finally becomes constant at, say, 200 deg. F. Assume that the condensing water cools the liquid ammonia to 80 deg. F. Then the pressure of the vapor mixture in the generator (and in the condenser) will be approximately that resulting from the sum of the partal pressures due to the vapor tension of the water vapor acting alone and the ammonia

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pressure acting alone. The pressure of the ammonia will be nearly that corresponding to a condensing temperature of 90 deg. F. or 181.8-lb. absolute added to the amount due to a water temperature of 200 deg. F., which will give an approximate steam pressure of 11.5-lb. absolute. The total pressure is about 193.3-lb. absolute, or 178.6-lb. gage.

So not only is there an increase of pressure on account of the distilling action, but steam itself will be generated (some 5 per cent) which mixes with the superheated ammonia and passes out of the generator. Provision must be made to remove as much of the steam as is possible before letting the ammonia gas pass into the condenser.

The result is that a water cooled-rectifier or dehydrator is used to reduce the temperature of the gases passing out of the generator. This device is simply a preliminary condenser but is operated to reduce the temperature only to within 10 to 20 deg. of the temperature of condensation of the ammonia.

ABSORPTION REFRIGERATING MACHINES

A sectional view of the Carbondale exhaust-steam refrigerating machine is shown in Fig. 90, and briefly described, the method of operation is as follows: The generator is filled with sufficient aqua ammonia to cover the steam coils, and the absorber with enough to submerge the water tubes. The brine pump is started and brine is circulated through the cooler coils. Then the cooling water is turned into the machine. The inlet water enters the bottom header of the condenser and, after it passes through the condenser, enters the absorber. It circulates through the tubes of the absorber; thence through the weak-liquor cooler, and finally into the rectifier. By this arrangement the water is used four times, each stage being at a somewhat higher temperature than the preceding. Thus the water consumption of the machine is greatly economized.

Steam is now gradually turned into the generator coils until the full exhaust-steam pressure is reached. As the steam heats the ammonia in the generator, the "generator" pressure, which indicates the pressure in the generator, con-

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denser and rectifier, will rise until it reaches a point sufficiently high to condense the ammonia gas in the condenser.

As the gas passes through the rectifier on its way to the condenser, the cool water circulating through the tubes of the rectifier chills the gas sufficiently to separate any entrained moisture that may be contained in it. The dry, or anhydrous, gas passes to the condenser, where it is condensed and falls in liquid form into the anhydrous receiver. The liquid anhydrous ammonia thus formed is then allowed to pass through the expansion valve into the brine cooler, the same as in the compression system. The expansion valve is throttled so as to keep a constant liquid level in the anhydrous receiver.

The pressure in the brine cooler is much below that of the condenser, and this drop in pressure causes the ammonia on entering the cooler to boil and absorb the heat from the brine in the coils. This changes it from liquid form on entering the cooler into a gaseous state on leaving it. The method of condensing the ammonia gas and the refrigeration that is produced in the brine cooler are identical with the compression system.

The problem now is to recover this gas. To do this, the weak ammonia liquor, which was left behind in the generator, and from which the gas has been expelled by the heat in the steam coils, is drawn from the bottom of the generator, through the exchanger, and thence through the weak liquor cooler into the absorber. Owing to the great affinity of water for ammonia gas, this weak ammonia liquor absorbs the gas in the absorber as it comes from the brine cooler, and by this means keeps down the pressure in the cooler. In turn, this weak liquor becomes enriched or strengthened by the ammonia gas and forms a strong liquor. This strong liquor is drawn from the bottom of the absorber by the ammonia pump, which discharges it into the exchanger, where it circulates around the coils that contain the weak liquor and then passes into the generator. The exchanger is simply a heat exchanger and answers the same purpose as a feed-water heater in a steam boiler plant. It heats the strong liquor on its way to the generator and cools the weak liquor. We

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now have the strong liquor back in the generator, ready for redistillation, and the cycle is complete.

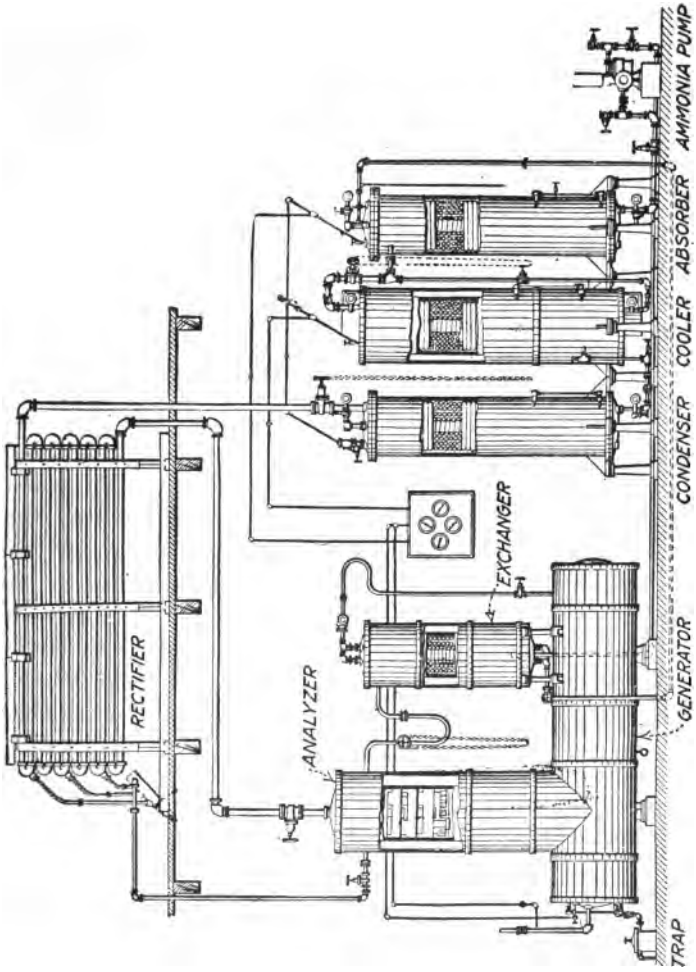


FIG. 91.—The shell absorption system.

Three types of absorption machines are in general use—the shell, the atmospheric and tubular types. The original Carbondale construction was of the shell and coil type shown in Fig. 91. This was a development of the old Pontifex design,

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which was brought unto this country in 1884, and has been gradually improved. This type is admirably suited for condensing water that is free from dirt and scale.

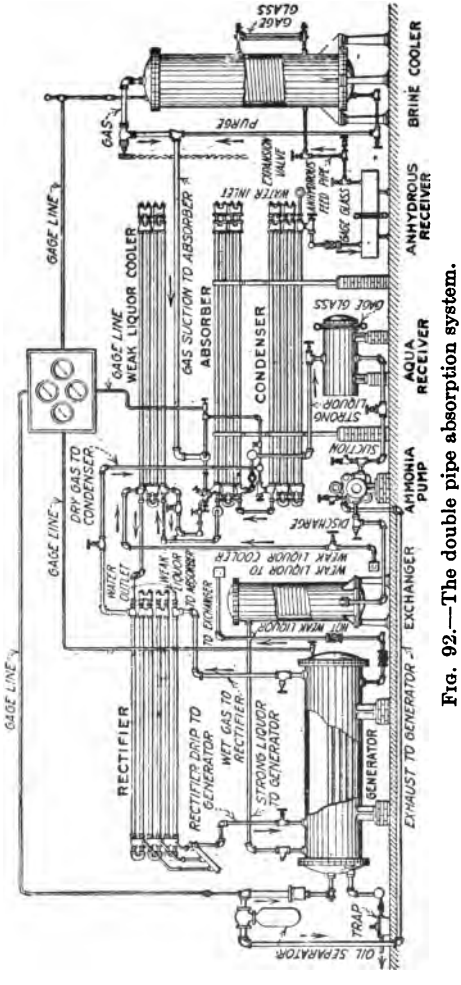


Fig. 92.—The double pipe absorption system.

In case of warm or muddy water the atmospheric type shown in Fig. 91 is recommended. The rectifier is placed under the weak-liquor cooler, so that the cooling water is used by this cooler before it descends to the rectifier. The condenser is placed above the absorber, so that the water passing over it runs down over the absorber, reducing the amount of condensing water required. Though a helical coil is shown, this construction can be used with expansion coils in the ice tank or with a tubular cooler, as preferred.

The tubular type of machine, Fig. 92, is the latest development, and consists of a generator similiar to the preceding types, and the equipment shown in the drawing.

The advantages of the absorption system are: (1) It takes very little steam at 30 lb. per hour per ton of refrigerating

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effect; (2) if exhaust steam is available it takes no live steam save for the ammonia pump; (3) it has no heavy moving parts; (4) it has no oil to clog and insulate the pipes; (5) it does not require heavy foundations, can be placed on the top floor of a building and makes no noise or vibration; (6) it can easily and economically produce low temperatures.

Operating and General Information.—To start the machine, assuming that the machine has a normal charge, proceed as follows: (1) Start water circulating through machine; (2) start brine pump, making sure that the brine is circulating through cooler; (3) turn steam gradually on the generator (this steam should never be entirely shut off); (4) open the gas line between the cooler and the absorber; (5) open the weak liquor valve setting it as in usual running; (6) start the ammonia pump; (7) when the generator pressure is raised to about that usually carried, slowly open the valve between the rectifier and condenser; (8) open the expansion valve and set it. The machine now is in regular operation.

To stop machine: (1) Shut off steam on the generator, leaving a little turned on; (2) shut the expansion valve; (3) shut down the ammonia pump; (4) close the weak liquor valve; (5) close the cooler and absorber gas line valve; (6) close the valve between the rectifier and condenser; (7) shut down brine pump; (8) shut off water supply.

The causes of lack of capacity are: (1) Insufficient water supply; (2) insufficient brine supply; (3) insufficient ammonia charge; (4) air or foul gas in the machine; (5) dirty condenser, absorber and rectifier coils; (6) cooler in need of purging.

Steam pressures: In case of the exhaust machine the steam pressure is usually in the neighborhood of 3 lb.; and does not vary. In case of the standard machine the steam pressure should be regulated according to the water temperature. Ordinarily, the steam pressure should never be in excess of one-third of the generator pressure. For example, with 150-lb. generator pressure, 50-lb. pressure on the generator is the maximum. If the machine is properly charged to do full capacity, a higher steam pressure will tend to dissociate the ammonia gas. Make sure that the generator steam coils are

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clear of condensation. An air cock is provided on the bottom header for the purpose of relieving air from the coils and also indicating whether any condensation backs up in the coils. At times traps do not work properly, and it is always well to keep this air cock open slightly to indicate any accumulation of condensation. When the machine is shut down, except for a long period, a little steam should always be allowed to enter the generator, so as to keep the shell warm.

Ammonia charge: With the machine in regular operation, and a normal charge, the following will be the conditions as indicated by the gage cocks on the various shells. The generator will have 1 to 2 in. of liquor covering the coils. The absorber will have 1 in. of liquor covering the tubes or, if a shell type or atmospheric machine, it will have about 6 in. of liquor in the shell or receiver. The condenser will have 4 to 6 in. of anhydrous in sight, and the brine cooler will have enough anhydrous in it to frost over both gage cocks and show a flush of soapy fluid in the glass when the cocks are opened. The gas line should be slightly frosted back to the absorber. This, however, should never frost sufficiently to show frost on the purge line.

Water supply to machine should be as regular as possible and a branch should be taken off the main at a point where other connections would not interefere with the flow to the machine.

Purging the cooler may have to be done as often as once a week, and then again not for 2 or 3 months. A cooler needs purging under the following conditions: (1) If both gage cocks are frosted, as with normal charge; (2) if the cooler pressure is only a little, if any, in excess of the brine temperature; (3) if, on opening the cooler gage, the liquor looks watery and not soapy, then the indication is that some aqua has worked over into the cooler and that the cooler needs purging.

To purge the cooler, close the expansion and gas-outlet valve and open the purge valve leading out of the bottom of the cooler. This will drain all the liquor out of the cooler into the absorber. The pressure will rise in the cooler at first, but after 15 min., will gradually fall. During the purg-

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ing the pump should be run slowly and the liquor be allowed to accumulate in the absorber, which will prevent the pump from losing its suction. When the cooler has been well drained out, open the expansion valve for about 30 sec. This will force the remaining liquor out of the cooler. After this is done two or three times and when the cooler pressure drops back to zero, the cooler has been completely purged. The machine can then be started, care being taken not to open the expansion valve too wide at the start.

The brine should preferably be made of chloride of calcium rather than from ordinary salt; it is more expensive than the latter, but does not have any corrosive action upon either pipes or valves and does not deposit solids in the coils. The gravity of the brine should occasionally be tested with the hydrometer. Brine of a specific gravity of 1.165 is safe at 6 deg. F. above zero; that of 1.200 does not freèze until it gets about 14 deg. F. below zero. If the brine is weak, there is liability of the cooler coils freezing, which would be indicated by a sudden stoppage of the brine pump. In such event immediately close the gas valve and open the expansion valve wide for at least five minutes. This should sufficiently heat the brine to thaw it and permit the pump to start.

To thaw freezer cooler, close the gas line from the cooler to the absorber and open the expansion valve wide and warm gas from the condenser will thaw out coils. Never allow the brine pump to stop when the cooler is in operation, as this is one of the causes of freezing.

When running at full capacity, the ammonia pump should circulate about $\frac{5}{8}$ gal. of liquor to the ton of refrigeration. If a steam pump does not run steadily, but with a jerk, it is probable that there is some difficulty with the valves. These are of hard rubber or steel and should be inspected occasionally. If they become pitted in the crossbars of the valve seats, it is best to replace them with new valves. Pump rods should be kept smooth and true; any difficulty experienced in properly packing the pump end can usually be traced directly to the fact that the rod has become worn, tapering from the middle to the ends, which has a tendency to burst the stuffing box by

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the wedge-like action of the rod. A properly packed box requires only a very light pressure of the gland and follower on the packing to prevent leakage. A small quantity of oil is to be used occasionally to lubricate the rod. If it is found that the pump is air-bound or gas-bound, it can be relieved by closing the suction and discharge valves and emptying the pump into a bucket of water through the pumping-out valve at the discharge of the pump. The suction valve should then be slowly opened and the pump allowed to fill with ammonia.

THE ABSORPTION REFRIGERATING MACHINE

Proper conditions of operation are necessary to good results in the use of both the compression and the absorption machine. Too high a suction pressure will destroy useful refrigerating effect for the compressor; for the absorbtion machine, unless the generator distills off the ammonia properly and the absorber provides the necessary *stronger* aqua, the concentration of the aqua in the generator will not be maintained, and poor refrigeration will result. With the compression machine it is only necessary to operate with a proper suction pressure to get the desired refrigeration, but with the absorption machine the aqua must be of the proper average strength, the steam pressure must be high enough, the absorber must give aqua of high enough strength and the aqua pump must be operated at a desirable speed. This seems complicated, but it really is not so, as will appear in what follows.

Suppose we have a generator containing aqua of 26 per cent (26 per cent by weight of ammonia and 74 per cent by weight of water) concentration. Suppose also that a condenser pressure of 161-lb. gage is carried (this is determined by the amount and temperature of the cooling water), such a solution at this temperature and condenser pressure will boil at 245.5 deg. F. This is similar to the action of heat on pure water. We say ordinarily that water in an open vessel will boil at 212 deg. F.; also that the temperature of the water (and steam) in a boiler at 125-lb. gage is 353.1 deg. F., and the temperature of steam under a vacuum of 26 in. is about 153.0 deg. F. In other

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words, when speaking of saturated steam, there is a definite, unvarying pressure for each particular temperature. This is also true for anhydrous ammonia, but aqua ammonia is different, because there is water present, varying in amount from 70 to 80 per cent. But if we have a certain concentration and a certain pressure, then the solution has a maximum concentration at a certain particular per cent. That means that at 26 per cent under the pressure of 161 lb. and a temperature of 245.5 deg. F., there could not be any more ammonia dissolved. In other words, a saturated concentration at 26 per cent and a condenser pressure of 161 lb. will have a temperature of 245.5 deg. But more important than this, if the pressure is 161 lb. and a temperature of 245.5 deg. F. is maintained, a concentration greater than 26 per cent will cause some ammonia to be driven off as a gas. If the solution was 27, 28, or 30 per cent, then while the temperature and pressure are maintained constant, a boiling action would continue until the remaining solution has a strength of 26 per cent. This is true of any solution of ammonia and water; that is, there is a certain fixed relationship of the pressure, the temperature and the saturation concentration, and if any two are known, the third may be found, as will be seen from Table XIII.

Such being the theory of the action of the generator, let us see how the action is accomplished. In the generator steam coils are provided, so arranged as to supply the heating surface necessary to drive off the ammonia gas and steam. This is somewhat similar in manner to the case of a steam boiler. In a water-tube boiler we usually say that 10 sq. ft. of heating surface is required per boiler horsepower. In the generator 14, 15, or 16 sq. ft. of heating surface is used per ton of refrigeration. Of course in the generator we could use very hot steam in the heating coils—say steam at 75-lb. or 100-lb. pressure—and in consequence of such intense heat (comparatively) the heating surface could be cut down to 12 or 10 or even to 8 sq. ft. Nowadays, however, the demand is only for low-pressure (exhaust) steam absorption machines. In the case cited it might be desirable to boil the solution down to 22 per cent (requiring a temperature of the aqua of 262.1 deg. F.), and as a mean con-

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TABLE XIII.—TEMPERATURE, PRESSURES AND CONCENTRATIONS FOR AQUA AMMONIA*

Pressure on aqua in pounds per square inch absolute

..4..	.10.	15	20	25	30	40	50	60	70	80	90	.100	.110.	.120.	.130.	.140.	.150.	.160.	.170.	.180.	.190.	.200.
2	169.0	189.4	206.5	220.0	213.1	249.4	264.3	277.9	288.8	299.0	307.8	316.2	323.7	330.7								
4	161.0	180.8	197.5	210.5	221.9	240.3	255.0	267.7	278.9	288.6	297.3	305.6	313.4	320.3								
6	153.0	172.3	188.8	201.5	212.4	229.5	244.0	258.0	169.0	278.5	287.4	295.3	302.6	309.6	316.0	322.2						
8	143.5	164.2	180.1	203.7	221.4	236.0	248.2	259.0	268.5	277.3	285.1	292.5	299.6	305.8	311.6							
10	137.4	154.3	171.1	183.7	194.6	212.3	227.0	239.6	250.5	259.5	267.9	275.8	282.9	289.4	295.7	301.8	307.4	312.6				
12	129.0	147.8	163.1	175.1	185.5	203.3	217.2	229.2	240.0	249.4	258.1	265.6	272.7	279.7	285.6	291.5	297.0	302.2				
14	120.5	140.1	155.5	167.4	177.6	195.2	209.1	220.8	231.2	240.5	248.9	256.1	263.0	269.7	276.0	281.8	287.3	292.4	297.1	302.0		
16	113.0	133.0	147.4	159.3	169.4	187.1	200.5	212.0	222.3	231.6	239.9	247.3	254.0	260.4	266.4	272.2	277.5	282.8	287.3	292.0		
18	107.5	124.9	139.4	151.4	161.2	178.3	192.0	203.1	213.5	222.6	230.9	238.0	244.7	251.0	257.0	262.5	268.0	273.0	277.6	282.4	287.0	291.7
20	99.0	117.3	132.0	143.6	153.8	170.2	183.6	195.4	205.0	213.8	221.8	229.1	235.9	242.0	247.9	253.5	258.6	263.6	268.4	273.3	278.0	282.7
22	92.0	109.9	124.6	139.9	146.0	162.7	175.9	187.0	205.6	213.5	220.6	227.1	133.4	239.3	244.7	250.0	254.9	259.3	263.9	268.5	273.0	273.0
24	86.0	103.0	117.0	129.5	139.2	155.2	168.7	180.0	190.0	198.4	206.0	213.0	219.2	225.5	231.3	236.7	241.9	246.7	251.0	255.5	260.0	264.5
26	79.2	96.3	110.6	122.1	131.9	148.2	161.0	172.1	182.0	190.7	198.2	205.1	211.5	217.6	223.4	228.7	233.7	238.4	243.0	247.2	251.5	255.8
28	73.4	90.6	103.6	115.3	125.0	140.6	154.0	165.2	174.7	182.8	190.3	197.1	203.1	209.3	215.0	220.4	225.5	230.3	234.5	238.9	243.2	247.5
30	66.5	83.8	97.0	108.5	117.8	133.5	146.1	157.0	166.4	174.9	182.7	189.4	195.6	201.3	206.9	212.2	217.3	222.0	226.7	231.1	235.1	239.3
32	61.0	76.9	90.0	101.0	111.0	127.1	139.4	149.7	158.8	167.3	175.0	182.0	188.2	194.0	198.4	204.5	209.4	213.8	217.7	222.0	226.4	230.8
34	55.0	71.6	84.0	94.8	104.5	120.0	132.0	142.4	151.9	160.2	167.8	174.6	180.9	186.6	191.9	197.0	201.7	206.1	210.0	214.2	218.3	322.5
36	49.0	65.5	78.5	88.8	98.1	113.5	125.8	136.0	145.3	153.3	160.7	167.3	173.5	179.3	184.7	189.7	194.5	199.1	202.9	206.8	210.6	214.4
38	43.0	60.0	72.5	83.2	91.8	106.8	119.4	130.0	138.7	146.7	153.8	160.3	166.4	172.2	177.5	182.5	187.3	191.7	195.6	199.3	203.0	206.9
40	37.0	53.8	66.0	77.3	86.0	101.0	112.9	123.4	132.3	140.0	147.0	153.4	159.4	165.0	170.5	175.5	180.1	184.6	188.9	192.8	196.5	200.1
42	31.6	48.7	61.0	71.5	70.4	94.0	106.3	116.4	124.9	132.5	139.8	146.4	152.7	158.2	163.5	168.5	173.3	177.7	181.4	185.3	189.2	193.0
44	27.5	43.2	55.3	65.3	73.6	87.8	100.1	110.3	118.8	126.3	133.4	139.9	146.0	151.5	156.7	161.6	166.3	170.8	174.3	178.2	182.0	185.0
46	21.2	37.6	49.7	59.6	68.1	82.0	94.0	104.2	112.8	120.1	126.9	133.4	139.5	145.0	150.2	155.0	159.4	163.8	167.5	171.3	175.2	179.9
48	16.0	32.0	44.4	54.2	62.7	76.7	88.3	98.5	106.7	114.0	120.6	127.0	132.9	138.4	143.7	148.6	153.3	157.5	161.2	164.9	168.6	172.5
50	11.0	26.8	38.9	48.4	56.8	70.9	82.6	92.6	100.8	108.3	114.9	121.0	126.9	123.4	137.2	142.0	146.4	150.7	154.9	158.7	162.4	166.0

*First column gives per cent concentration; figures in columns are boiling temperatures for the pressure at top of column.

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centration of 26 per cent would be used, then 30 per cent concentration could be looked for in the strong aqua. The steam pressure must be such as to give a sufficient temperature difference between the steam in the coils and the aqua. For example, 262.1 + 10 deg. = 272.1; that is, a 28-lb. steam pressure is required.

The action then of the generator is to boil off a vapor which is a slightly superheated ammonia. This is obtained by having a temperature sufficiently great in the steam coils, determined by means of the values given in the table. Increasing the steam pressure or decreasing the head pressure or increasing the concentration will stimulate the boiling action, but the easiest means of control in the hands of the operator is the steam pressure. If capacity is not being obtained, a rise of steam pressure will give hotter steam, therefore a greater temperature difference and a greater heat transfer. The result will be that more steam will be condensed in the coils and more ammonia gas will be distilled out of the generator.

It is readily seen from the foregoing that one of two things in the generator may be wrong if the plant is found not to condense the ammonia properly—either there is too weak a concentration in the generator or the steam pressure is too low. In either case charging more anhydrous into the system or increasing the steam pressure will give the same results;

TABLE XIV.—POUNDS OF STRONG AQUA REQUIRED PER POUND OF AMMONIA EVAPORATED

Concentration weak aqua, per cent	Concentration—strong aqua							
	20 per cent	22 per cent	24 per cent	26 per cent	28 per cent	30 per cent	32 per cent	34 per cent
18	41	20⅛	13⅜	10¼	8.2	6.84	5.86	5.12
20	..	40	20	13⅛	10	8.0	6.77	5.71
22	..	....	39	19½	13	8.75	7.8	6.5
24	..	....	....	38	19	12⅜	9½	7⅜
26	..	....	....	....	37	18½	13⅛	9.25
28	..	....	....	....	....	36	18	13.0

namely, having enough difference between the condensing temperature of the steam and the boiling temperature of the

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aqua to enable sufficient heat to pass through the heating surface of the steam coils. All of this may be seen from the table. For instance, suppose it is found that ammonia is not being condensed fast enough in the condenser, though the latter is working properly and the rectifier is not cooling the gas too much. Suppose the steam temperature is 250 deg. F. and the condenser pressure is 161 lb. It is possible to take a sample of the weak aqua and test it for its concentration. This, however, is not really necessary if we have a means of measuring the temperature of the aqua as it leaves the bottom of the generator. Suppose the temperature is 245 deg. F. Referring to the table, the concentration is found to be only 26 per cent. If this concentration had been 28, then it is evident that boiling of the ammonia would have occurred at a lower temperature (237.1 deg. F.), more pounds of ammonia would have been evaporated. The same result necessarily would have occurred with an increase of steam pressure.

The action of the absorber is somewhat like that of the jet condenser, and reference to a few of these similarities may make the subject clearer. In a steam condenser the vacuum (or back pressure on the turbine) is produced by condensing the vapor. The total pressure exerted is the sum of the vapor (steam) pressure and the air pressure. This air pressure accumulates and has to be removed by means of an air pump, or the vacuum would be lost. In a similar manner the back pressure or boiling pressure of the ammonia in the brine cooler on the expansion piping is determined by the action in the absorber. The slightly superheated gas from the expansion coils passes into the absorber and is allowed to come in contact with a spray of weak cold aqua, or is allowed to bubble up in the aqua. Since the aqua is not pumped away as the water is in the jet condenser, the heat of condensation and of solution has to be taken away by an outside medium, which in this case is the cooling water used. But, as in the case of the surface or jet condenser, the total pressure in the absorber is equal to the pressure due to the vapor tension plus that due to air or inert gases present.

The vapor tension in the absorber is determined by the

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temperature and concentration of the strong aqua in a manner similar to that used in the case of the generator. For example, if the strong aqua is 90 deg. F. and 32 per cent aqua is being carried, then the vapor tension is (from the tables) 20-lb. absolute, or 5-lb. gage. But this 5-lb. gage is possible only when the absorber is purged out. A new plant is likely to have a large amount of air in the system, and this will slowly accumulate in the absorber. If air collects, which will itself exert a pressure of 3 lb. per square inch, then the total pressure in the absorber will be 8 lb. The pressure at which boiling of the ammonia takes place is not the absorber pressure. It tends to become this pressure, but in fact there is usually considerable "wiredrawing" between the two pieces of apparatus, and the result is that the pressures of the brine cooler or expansion headers are usually from 2 to 5 lb. in excess of the absorber pressure. This last loss of pressure may be eliminated by making the pipe-line connections and gas headers of liberal size. The inert gas may be removed in a manner similar to that used in the ammonia condenser.

Effect of Temperature of Cooling Water on Operation of the Plant.—Reference to the table also shows the effect of the temperature of the cooling water on the operation of the plant. For instance, the absorption machine finds its paritcular sphere of action in low-temperature work. A strong aqua of 32 per cent concentration and at 5-lb. gage must be cooled at least as low as 90 deg. F. If the temperature of the water rises so that the aqua is kept at 97 deg. F., then the aqua concentration (see table) will be 30 per cent instead of 32. Usually, the absorption machine operates under small differences of concentration, like 22 per cent to 26 per cent. If the machine had been designed for operation between 26 and 32 per cent and the aqua fell in concentration to 30 per cent, the aqua pump would have to be speeded up and the steam economy in the generator (as well as the economy of water used in the absorber) would be decreased.

The reason for the last statement may be seen on examining the action of the strong-aqua pump. This pump has a function similar to the boiler-feed pump which takes the feed

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from an open feed-water heater. But the strong aqua is only a carrier—1 lb. of strong aqua has a little (sometimes a very little) more ammonia absorbed in it than does the weak aqua entering the absorber. This strong aqua carries back to the generator the ammonia absorbed in the absorber. Therefore, if it takes 12⅓ lb. of strong aqua to carry back 1 lb. of ammonia, a drop in concentration from 32 to 30 per cent would necessitate pumping 18⅔ lb. of aqua, or one-half more than before. So, evidently, it is important to keep the strong aqua as strong as possible, and for satisfactory operation of the plant a means of getting the strong-aqua temperature as it leaves the absorber should be provided. If no inert gases are present, then the concentration may be obtained from the aqua tables. The effect of the concentration on the action of the pump may be seen from Table XIV.

Generally speaking, it is wise to keep the difference in the concentration as great as possible. The reason for this is that the weak aqua has to be cooled by the cooling water before admitting it to the absorber, and also the strong aqua has to be returned to the generator at a temperature much below the temperature of the aqua in the generator.

In conclusion, it is recommended that the operating engineer run the machine by reference to a table on the properties of aqua ammonia.

ABSORPTION-PLANT TROUBLES

In one obsorption plant of moderate size, after several years of steady operation, the generator was thoroughly overhauled. Both heads were off and new gaskets were put in when the heads were replaced. Needless to say, the header bolts were made as tight as it was possible to make them, and we were of the opinion that they were tight for good. The generator ends were then covered up, magnesia packed around them, and they were forgotten.

About a year after the heads were covered a leak developed somewhere around the plant, and though it produced quite a "smell," it could not be located. Finally one of the hunters

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put his nose to a joint in the jacketing near one end of the generator and got a "shock" that sent him reeling backward. That meant that the covering would have to come off to get at the header bolts. On taking off the cover some of the nuts were found to be only hand tight. The header bolts on both ends of the generator were found to be in the same condition.

It is more of a job to take off and replace the cover in a task of this nature than it is to tighten up the bolts. Also the removal and replacing of the magnesia around the joint necessitates making more or less of a dirty mess each time the operation is performed. The covering is still off the generator ends. When it is replaced next time an effort will be made to devise a way of putting it on so that it may be easily removed. The generator jacketing should be so constructed that the removal and replacement of the part that covers the header bolts would be a matter of only a few minutes work. Then these bolts could be inspected at frequent intervals and tightened whenever necessary.

ABSORPTION-SYSTEM TROUBLE

One evening the ammonia pump developed signs of distress that called for an investigation. The first thing noticed was that the ammonia pump was running slower than usual, the throttle valve having the customary opening. The pump was given more steam, but this only brought it up to speed again for a short while after which it slowed down aeain. The throttle valve was opened wider and wider until it was nearly fully open, but even then the pump would not run fast enough to do its customary work. It acted as though the exhaust-steam passages leading from the pump had become partly closed, making the pump work against high back pressure. Examination of all the valves on the exhaust side of the pump did not disclose any out of the ordinary condition, and it was a question what to try next.

Going back to the pump, the operator opened the throttle wide and waited to see what would happen. The pump speeded up a little and then slowed down again. A hissing sound from the direction of the exchanger then attracted

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the operator's attention, and he looked over that way. One look was enough. He saw streams of ammonia liquor spurting out around the top joint of the exchanger at every stroke of the pump. Apparently the trouble was in the ammonia side, and later events proved that to be the case. A piece of waste, probably left in the machine when it was overhauled, had worked its way through the pipe connecting the exchanger and

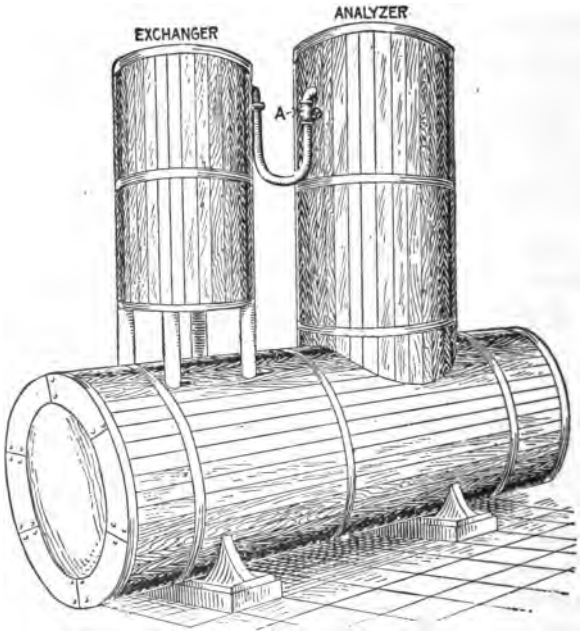


FIG. 93.—Choked absorption system.

analyzer, Fig 93, until it came to the valve *A*, where it lodged. When the pipe and valve were taken down for examination, the waste was found wedged in the valve chamber.

Since the foregoing incident, computation has shown that there must have been between 700- and 800-lb. pressure per square inch on the exchanger and the pipe connecting the exchanger and analyzer, when the top head of the exchanger started to leak. No wonder the header bolts stretched!

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UNPACKING AND REPACKING THE AMMONIA PUMP

Packing the ammonia stuffing box on the pump is a job that the absorption-plant operator must do at more or less regular intervals. Like other disagreeable jobs, there is a way to do it that reduces the unpleasantness to a minimum.

The layout that will be used as an illustration is planned as shown in Fig. 94. It will be assumed that the plant has been shut down for a few hours and that packing the pump is the next thing in order. Valves 1, 4 and 5 are opened and the liquor is pumped out of the absorber, after which the pressure

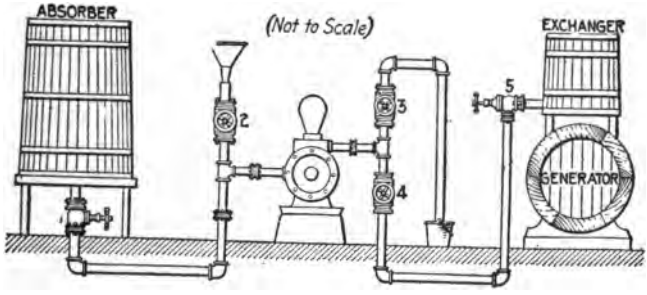


FIG. 94.—Packing ammonia pump.

in the absorber should be at or near zero. Valve No. 1 is then closed, and if the liquor in the machine is not already too "weak"—that is, if the percentage of ammonia is not less than it should be—a couple of bucketfuls of distilled water is put into the funnel (shown just above valve No. 2) and forced into the generator by the pump. Valves 4 and 5 should then be closed and No. 3 opened, and either by hose connection or bucket, tap water should be fed into the funnel and the pump allowed to force it through valve No. 3 into another bucket or, better still, a connection to the sewer. This should be continued until the pump chambers are thoroughly flushed out. Then, without shutting off the water and with the pump still running slowly, the stuffing-box gland should be eased off. If the pump is allowed to run slowly, the gland may be entirely unscrewed and the water will force out all the old packing without injury to the gland or spool. This is a much

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easier, cleaner, quicker and less odoriferous method than when the packing is pulled out of the box dry with a packing hook.

Having now unpacked the pump, proceed to repack it. Consider Fig. 101. The stuffing box (this particular one) is long enough to hold about 12 rings of packing; 9 of them are put in ahead of the spool and 3 behind it. There is an enlargement in the lower side of the stuffing box and into this section two plugs, *A* and *B*, are screwed. Running lengthwise in this enlarged section is a port which is a continuation of the cavity filled by plug *A*; this port empties into the suction-valve chamber. The cavity filled by plug *B* is continued as a port

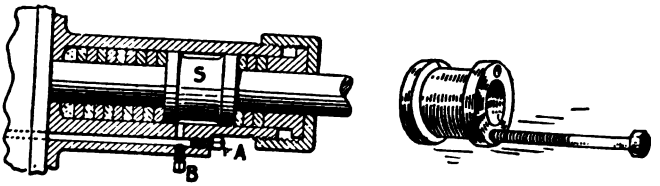


FIG. 95.—Stuffing box lantern.

up through the lower wall of the stuffing box. The spool *S* is shown separately in Fig. 95. The view given is that of the bottom of the spool as it lies in working position in the box. There is a port, not shown, almost the full length ofthe spool; on the crosshead end there are two ⅜-in. holes tapped an inch or so deep.

With the box clean and the spool and plug *B* removed insert the first ring and push it in as far as possible with the fingers, then farther with the spool. Now take a ⅜-in. bolt about 12 in. long and screw it into the spool a couple of threads. With the bolt for a handle, use the spool for a ram and force the ring back hard against the bottom of the box. Follow this ring with another, staggering the joints in the packing. Continue this until the end of the long port in the bottom of the spool just overlaps the port extending up from plug *B*. This is determined by running a small bolt or pencil up into the port. When this condition obtains, there is enough packing behind the spool, and leaving it in this position, enough rings

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are put in after it to fill up the box. Replacing the gland and plug *B* and closing valves Nos. 2 and 3 completes the job.

The reason for the ports extending from plugs *A* and *B* are thus explained: Any ammonia liquor that finds its way through the packing to the spool is returned to the suction valve chamber through these ports. Taking up on the gland forces the spool farther into the box, but the shape of the spool allows the liqour to find its way to the return ports no matter what its lengthwise position in the box. It is well when repacking the pump to remove plug *A* also and see that the longitudinal port is clear all the way. Bits of packing are apt to lodge there and obstruct the flow.

It might be well to tell at this time of the disagreeable experience of an engineer who started a plant of this kind shortly after the pump had been repacked. He knew that the pump had been worked on and thought that everything was all right for starting. He proceeded to open valve No. 1 and had only got it started when a stream of ammonia liquor gushed from the funnel and came down on his head and back. Fortunately, his face was turned away from the funnel. There were other men working in the next room and they helped him to the water tap and showered his head with water. Investigation showed that valve No. 3 had not been closed when the job of repacking the pump was completed. The high pressure in the generator had forced some liquor over into the absorber through imperfectly seated valves in the weak-liquor line. Some pressure had thus been built up in the absorber, and when he opened valve No. 1, there was nothing to prevent the ammonia spurting out of the funnel. Luckily, he suffered no serious injury, but the experience taught him, and also those who were responsible for the accident, to be more careful in the future.

LIFE OF AMMONIA-PUMP RODS

As is generally known, the life of the piston rods of ammonia pumps is comparatively short, and the rods must be packed quite frequently to prevent leakage of ammonia. This frequent packing wears the rod considerably, and many engineers

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keep on hand several sizes of packing rings, so that as the rod wears smaller the packing rings used are correspondingly smaller, to make a tight joint. Keeping on hand various sizes of packing rings and repacking the rod every 2 or 3 weeks is a source of loss of ammonia.

Inquiries of 230 operators in one year showed that 25 had to renew the rod every 3 months and had to repack it every 3 weeks. This was in 24-hr. service, although most of the plants running 18-hr. experienced the same trouble. Correspondence with various ammonia-pump manufacturers regarding the extreme wear of the rod corroborated the experience of the operators.

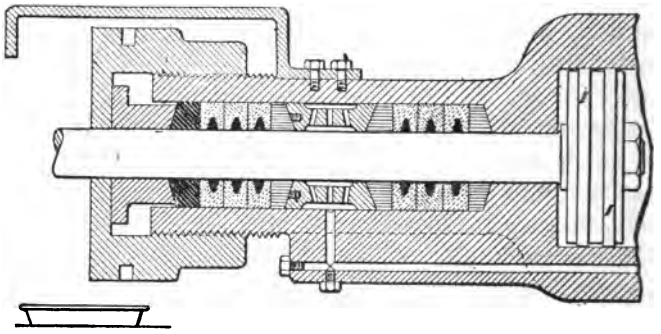


FIG. 96.—Ammonia pump rod.

A refrigerating engineer who has done considerable experimenting with pump rods of various materials states that a high-grade tool steel gives the best service. In one particular instance a local machinist made one of these rods without annealing it. This rod was packed with fibrous packing having a metal or babbitt insertion, the stuffing box being packed as shown in Fig. 96.

This rod has been in service for more than a year, running 18 hr. a day in summer, and 6 to 8 hr. a day in winter, and the wear is only 0.002 in. The gland has been taken up approximately $\frac{5}{16}$ in. in all this time, and the rod is operated at about blood heat. From 5 to 6 drops of oil per minute is fed over the rod, and it is only when starting that one gets a slight odor

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of ammonia, say until after the second or third stroke is made. From then on the odor of ammonia is not noticeable.

In packing the rod it is advisable to put standard ammonia packing rings in the bottom of the box and then alternate with the metallic rings, or rather the fibrous rings having metallic insertions. The last ring to be put on should be of a good resilient rubber, placed next to the gland where it can take up expansion and contraction.

ABSORPTION-MACHINE AMMONIA LEAKS

Generally speaking, the oftener a machine is started and stopped the more likely it will be to develop leaky joints. A joint that is tight when the machine is hot and in operation may leak when the machine is shut down and comparatively cold. For this reason intermittent operation of absorption plants should be avoided as much as possible.

The joints on those parts of the machine subject to the greatest variation in temperature and pressure are the ones most likely to require frequent attention. Among these are the stuffing boxes around the coils protruding from the end of the generator and also those on top of the exchanger. It is a good plan to go over the bolts on such apparatus shortly after the machine is stopped and again at frequent intervals while the generator is cooling. Care should be taken not to squeeze any harder on the packing than is necessary to keep the joint from leaking. Unnecessary tension on the packing simply shortens its life and hastens the time when the joint must be repacked.

A few years ago it was the practice to pack the stuffing boxes mentioned with lead-inserted round rubber packing. This came in long sections which required cutting when being applied. Present-day practice is to use solid rubber packing which comes in ring form. It is molded to the proper size and must be cut before it can be put in place. The cuts should be made diagonally as usual and staggered to obviate as far as possible any leakage through them.

When getting ready to pack the generator and exchanger coil boxes, the liquor in these shells should be transferred to another part of the machine and the pressure reduced as far as

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possible. With the generator under pressure, opening the weak-liquor valves will allow the bulk of the liquid in the generator to flow into the absorber and cooler. The next step is to get some cooling water through the generator steam coils. In some machines provision for this is made by drilling and tapping the upper steam header for a ¾-in. or larger plug, which is removed when it is desired to run water through the coils and a hose or pipe connection made to the plug hole. Running cool water through the coils will tend to condense what ammonia vapor there is in the shells and thus reduce the pressure. It is not a good plan to try to remove all the old packing, but the first three or four rings around each coil; this is because it is not possible to remove all the liquor from the generator by means of the weak-liquor line and there will always be a little in the bottom, the vapor from which would be extremely disagreeable, even if not dangerous, to one taking the old packing from the bottom of the box.

Next to the joints already mentioned, the most common sources of leakage are the stuffing boxes on the generator and condenser gage-glass valves in the weak liquor line, and the expansion valve. It is well to pack these boxes at the same time the packing in the generator and exchanger coils is being renewed, even if they apparently do not leak; doing it then may save time and trouble later on. The gage-glass boxes on the absorber and cooler require little attention in that respect because those shells are worked under low pressure.

The header joints on the ends of the generator are good places to watch for leaks. The only thing that can be done if a leak develops there is to take up on the header bolts. When that fails, a new gasket is the only remedy. If the gasket on one end is renewed, renew the other.

The senses of sight and smell are usually all that the experienced operator requires in detecting leaks from joints exposed to the air, although litmus paper, and sticks of sulphur as indicators are used to advantage. A different class of leaks is that where the joints are directly in contact with water or brine. To ferret out leaks of this character, the use of either litmus paper or Nessler's solution is necessary.

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OBSTRUCTIONS IN THE WEAK LIQUOR LINE

Everyone appreciates that the temperature of the cooling water for a refrigeration plant influences its operation and output considerably—especially so in the absorption system. In Texas and other parts of the Southwest the water is frequently extremely hot for months at a time. The particular plant under discussion—a 20-ton absorption system with atmospheric condensers and absorbers—formerly took water from a well, which later developed sand trouble, decreasing its capacity. The cooling tower was much overworked, and the temperature of the water in the supply tank under the tower on this particular day stood at 95 deg. Trouble started when the engineer began to purge the 25-ton brine coolers. This caused the absorber pressure to rise to about 30 lb., and immediately the generator boiled over, there being perhaps a leaky check valve which allowed weak liquor to pass into the coolers. The steam pressure in the generator was 35 lb., but there was not a drop of weak liquor showing in the gage glass. In about 5 min. the absorber pressure rose to 70 lb., and was still going up. The generator pressure rose to 280 lb., and the culmination of events took place so suddenly that the engineer lost his head.

The steam was shut off the generator, and the weak-liquor valve to the absorber was closed, as well as the gas valves from the two vertical brine coolers. Then the relief valves on the generator steam line were opened to the atmosphere, and the speed of the water pump increased. These done, the pressures gradually decreased throughout the system, but there were no indications that the ammonia pump was getting liquor.

When the absorber pressure reached 65 lb., it did not drop further, although plenty of water was going over the absorbers. This pressure should have gone down much more rapidly. That it did not showed that something was wrong somewhere. There seemed to be no weak liquor in the generator, and the pressure in the brine coolers was the same as that in the absorber, which showed that flow of gas between the coolers and the absorbers was not taking place. The engineer shut

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off two sets of absorbers and allowed one set to remain in service. Into this he turned weak liquor, and the pressure began to drop rapidly. The ammonia pump was started, and in 4 min. the absorber pressure dropped to 13 lb.

Believing that he had overcome the trouble, he cut in one more coil and found that there was 2 in. of liquor in the generator gage glass. He cut in No. 3 set of coils, and almost immediately the pressure rose to 50 lb. and the liquor disappeared in the glass. The pressure, instead of reducing, slowly rose. The faulty coil was cut out, the pressure

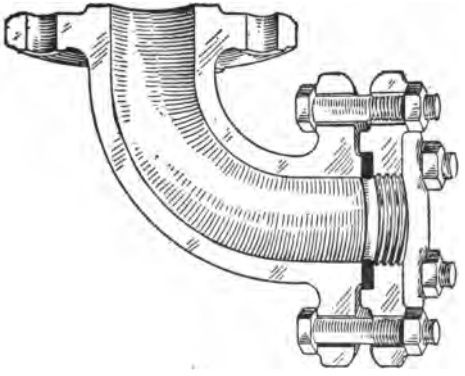


FIG. 97.—Faulty ammonia fitting.

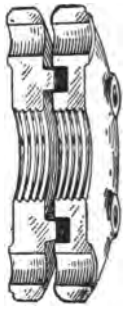


FIG. 98.—Good design of joint.

began to drop, and there was a rush of liquor to the pump, which was kept running until 7 in. of liquor showed in the generator gage glass. After about ½ hr. spent in purging the coils, the liquor level was back at about normal. On removing the aqua ammonia from the faulty absorber coil, all header valves were shut and the pressure was allowed to escape through the purge valve. The ½-in. plug in the line near the injector was removed and the line tried for freedom from obstructions by running a wire through the nozzle, but the wire could not be pushed through.

After disconnecting the injector header, a piece of rubber about the size of a large pea was found to be plugged tightly in the nozzle opening, forced there by the generator pressure.

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The nozzle cleared and everything put in shape, it was not long before the temperature was back to normal, and the engineer was congratulating himself on having so quickly remedied the trouble. The type of joint used let the rubber into the line (see Fig. 97). If a male and female joint (Fig. 98) had been used it would have prevented this trouble.

But just at this time the absorber pressure began to rise again and the liquid level to drop. The engineer ran to the absorber and felt all of the weak-liquor headers to individual absorber coils to locate the defective one, which was found, it being cooler than the others, showing that no weak liquor was flowing into the coolest coil. This was the one that was evidently causing the trouble. He reasoned it out this way: When the set of coils became gas-bound, it automatically made the others gas-bound, the check valves on the coolers closed, and the weak liquor kept on flowing into the absorber coils, lowering the liquid level in the generator, and there being no overflow to the aqua ammonia receiver, the absorber pressure rose. Being determined to find the root of the trouble, the engineer removed the weak-liquor header and cleared it thoroughly. Many pieces of rubber were found, which had come from the gaskets in the piping system.

The next day the engineer could not get a flow of gas from the coolers to the absorber. On investing the header was found to be plugged with rubber. This would indicate that strainers should be used on the pipe lines.

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CHAPTER IX

OPERATING THE CARBON-DIOXIDE REFRIGERATING MACHINE

In principle the carbonic system of refrigeration is similar to the ammonia wet-gas compression system. The cycle of operation is identical; high-pressure gas leaving the compressor is conducted to condensers, which may be of the double-pipe, atmospheric or submerged type. Here the gas is liquefied and passes out through a liquid receiver and liquid line to the expansion valve. At this point the pressure is reduced to the expanding (evaporating) or suction pressure and the liquid boils and absorbs heat in its passage through the expansion coils. It is necessary to bring back a saturated gas to the compressor to absorb part of the heat of compression, and a good rule to follow is to keep the discharge line at a little better than blood heat—just hot enough so it will not burn the hand when grasped. This temperature is of course controlled by regulating the expansion valve.

As the pressures in a CO_2 system are high, care must be exercised in making joints, and a good grade of extra-heavy iron pipe should be used for the entire gas system. The type of joint shown in Fig. 98 is generally used and, when properly made, is satisfactory. Pipe is screwed clear through the flange so that it projects beyond the face; the end of the pipe is then faced off flush with the face of the flange. The flange should then be screwed down about one-eighth turn, which leaves the pipe protruding slightly beyond the face of the flange. The joint is made on the end of the pipe, and pressure does not get to the threads. A fiber ring is the most satisfactory gasket to use, though a soft copper ring gives good results. Care should be taken to draw up the joints evenly at all four points, and it should be followed up occasionally during the

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first few weeks, operation. Pipes are generally bent instead of using ell fittings, and it is of advantage to use full lengths of pipe wherever possible, to minimize the number of joints. When pumping up air pressure to test for leaks on a new system, care should be taken not to allow the compressor to overheat. If the temperature in the pipes should reach the flashpoint of the lubricating oil, an explosion might result. Injecting part of a drum of CO_2 into the system while testing will remove this danger, as CO_2 will not support combustion.

The joints should be gone over with soapsuds until all leaks are located and stopped, after which the air can be blown out of the system and a drum of CO_2 connected for charging.

Start water on the condensers and see that the force-feed pump for the stuffing-box and cylinder lubrication is working properly. The machine can now be started and gas allowed to flow into the system. After pressure on the high side reaches that due to the temperature of the condenser water the expansion valve should be adjusted to regulate the pressure on the expansion coils. As rooms or brine tanks are naturally warm, a high back pressure, 30 to 35 atmospheres (450 to 525 lb.) should be carried at first to avoid overheating the compressor, and this can be reduced gradually as the brine or the room temperatures fall. In charging large systems it is sometimes necessary to stop the machine two or three times to allow it to cool.

In operating a CO_2 plant under normal conditions, the discharge temperature should be carried 15 deg. F. higher than that due to the temperature of the injection water (see table). For instance, if the injection water is 70 deg. F., the pressure in atmospheres on the condenser will be $70 + 15 = 85$ deg. or 73 atmospheres (1073.1 lb.) when the plant is fully charged and any drop of pressure, unless caused by a change of the condenser water or the setting of the expansion valve, would indicate a leak in the system.

In general practice a sufficient quantity of water is allowed to flow through the condenser to get a 10- to 15-deg. rise between the injection water and the discharge water. Leaks should be carefully watched for during the first few days'

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operation. Soapsuds may be painted on the joints; a lighted candle held around a joint will be instantly extinguished by even a small CO_2 leak. An ounce of oil of peppermint may be injected into the system and circulated around with the gas; a leak can then be located by the odor of the peppermint.

After operating a new plant about one week, the scale trap on the suction line should be cleaned. Shut all return valves on the expansion coils, run the compressor a few minutes until the low-pressure gage drops to zero; then the trap may be opened, the strainer removed and cleaned. As there is always considerable dirt and scale in a new system, it is important to clean this trap after first starting; after the first cleaning, once a year will be sufficient.

The oil trap should be blown out daily and the oil filtered. It may be used over again if desired.

If difficulty is experienced in getting proper results from a CO_2 system, it may be due to one of the following reasons:

1. Shortage of gas in the system. Keep the condenser temperature 15 deg. F. higher than the temperature of the injection water, and if the temperature is below this point, add gas to build it up. If the pressure keeps gradually dropping, it is certain that there are leaks in the system, and these should be found and corrected.

2. Water in gas Occasionally a small quantity of water is found in the gas drums, and this will cause trouble in the system by clogging the expansion valves and the compressor valves. Turn the drum upside down and open the valve slightly; if water is present, it will come out first. Manufacturers of carbonic gas are now taking pains to make a dry gas for refrigerating use, and they should be informed that gas is desired for use in a refrigerating machine, when CO_2 is ordered.

DETECTING LEAKY SUCTION VALVES

3. Leaking compressor valves. Valves sometimes become pitted by small particles of scale or dirt in a new system, and occasionally one valve will stick. A leaky or sticking valve at one end can be detected by the uneven vibrations of the hands on gages. Feel the caps on the discharge valves, and

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the hottest is the one that is leaking. A leaky suction valve can be detected by listening carefully with the ear close to the cap. The gas can be heard hissing through the leaky valve on the discharge stroke. When the valves are removed for inspection, see that both valve fibers have an even impression; if one is lighter than the other, add paper washers until they equalize. If the valve is pitted badly, it should be ground.

4. Worn or defective rod or piston packing. The leading manufacturers of CO_2 compressors now use an excellent type of cast-iron ring on the pistons, and these require no attention beyond an occasional inspection, say about once a year. The older machines, however, and some of foreign make use cup leathers on the pistons, and these should be renewed about once every six months.

When the machine shows a decreased capacity and all other conditions are normal, the piston packing should be examined and new leathers applied if necessary. The rod packing should be renewed about once a year, unless on examination it appears to be in good condition. Instances have been known where a set of ordinary rod packing has lasted several years. A set consists of four leathers held in brass keepers and several rings of good spiral packing in the outer box. The leathers are sealed with oil under pressure, which is supplied by an eccentric-driven pump. The outer box of spiral packing holds the oil pressure alone, and any leakage at the front gland will be oil, not gas. It is customary to keep a very slight drip at the front gland.

5. Dirty condenser tubes. Where dirty water is used, the condenser tubes will become lined or covered with mud or soft scale and prevent a free transmission of heat from the gas to the circulating water. When this condition exists the head pressure will be unusually high and the liquid-line temperature will be much higher than that of the injection water. Coils or tubes should be cleaned with a wire brush.

PIPE JOINT FOR CO_2 SYSTEM

Any good ice-machine lubricating oil with a low congealing point will be found satisfactory for use with a CO_2 machine.

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As CO_2 has no affinity for oil, the latter does not mingle with the charge, and after the system is clean, oil will be drawn from the trap clear and unchanged. The carbonic system is economical in the use of oil.

CORRESPONDING TEMPERATURES AND PRESSURES OF SATURATED CARBONIC ANHYDRIDE (CO_2)

Temperature, degrees Fahrenheit	Pressure, atmospheres	Pressure, pounds	Temperature, degrees Fahrenheit	Pressure, atmospheres	Pressure, pounds
−15	16	252.2	50	46.5	683.55
−10	18	264.6	55	49.5	727.65
− 5	20	294.0	60	52.8	776.16
0	22.2	326.34	65	57	837.9
5	24	352.8	70	61	896.7
10	26.3	386.61	75	65	955.5
15	28.2	414.54	80	69	1,014.3
20	30	441.0	85	73	1,073.1
25	32.5	477.5	90	77	1,131.9
30	34.3	504.21	95	82	1,205.4
35	37.2	546.84	100	86.2	1,267.14
40	40	588.0	105	91.5	1,345.05
45	43.5	639.45	110	96.8	1,422.96

To the refrigerating engineer who has never operated a CO_2 plant, an agreeable experience is in store. These machines, as built by the leading manufacturers, are so safe that a deliberate attempt to do damage with them would result in failure. If the entire charge should blow out in an average engine room, no discomfort would be experienced by the occupants; CO_2 is neutral and harmless to all metals, and will not injure foodstuffs. It is also a positive fire extinguisher. When it is desired to open the system at any point, the work can be started and finished promptly without running to the door every little while for fresh air as is so frequently necessary in plants using ammonia.

The cost of liquid CO_2 in drums is \$5 to \$6 per 100 lb., while anhydrous ammonia costs \$25 to \$30. As a rule more CO_2 is lost through leakage than ammonia, considering plants of equal efficiency, because CO_2 being odorless, a slight leak is more likely to be overlooked. This condition is offset, however, by the fact that a charge of ammonia, under

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the usual conditions will decompose to some extent and form non-condensable gases which must be removed occasionally by purging, while CO_2 is so staple a compound that it cannot decompose in the system and purging is never required.

The CO_2 cylinder being only one-fifth the size of the ammonia cylinder on machines of equal capacity, the total clearance volume can be kept very small, and as the ratio of expansion is only about three to one, high compressor efficiency is obtained and the loss from reexpanded gases reduced to the minimum.

Theoretically, the critical temperature of CO_2 is 90 deg. F., but this does not follow in practice, however in southern Texas, where the injection water was 90 deg. F., and owing to an insufficient supply, the discharge often runs at an average of 110 deg. F., but no difficulty was experienced in getting rated capacity from the machine.

For low-temperature work the CO_2 machine can reach points impossible with any other system; one machine installed in a chemical works maintains a temperature of 80 deg. below zero F. It is also especially suitable for air-cooling purposes in theaters, banquet halls, restaurants, etc., as the direct-expansion system of air-coolers can be used, thus avoiding the complicated system of brine coolers, tanks and pumps necessary with any apparatus using a poisonous or odorous gas as a refrigerant. Then, too, there is no fire hazard with CO_2.

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CHAPTER X

TESTING COMPRESSION PLANTS

It is the purpose here to relate briefly the essentials of the test of a refrigerating plant. For the present, the compression system only will be considered. Although they are more or less familiar, it is well to begin with definitions of the usual quantities sought, as follows:

Refrigerating effect is the amount of heat abstracted by the ammonia from the cooling medium, as brine, expressed in B.t.u. per unit of time (minute, hour or 24-hr. day). This includes the waste cooling effect due to heat transferred from bodies it is not desired to cool—a waste necessarily ensuing from imperfect insulation, manipulation, etc. *The unit of refrigeration* is a refrigerating effect of 288,000 B.t.u. per 24 hr., or 200 B.t.u. per minute. To obtain this effect by ice at 32 deg. F., it would be necessary to melt 1 ton (2,000 lb.) of it in 24 hr., the latent heat of fusion of ice being 144 B.t.u.

The capacity of a plant equals the refrigeration it delivers: that is, the number of tons of 32-deg. ice it would be necessary to melt per 24-hr. day to produce an equivalent effect. It is often termed the *ice-melting capacity.*

Ice-making capacity is sometimes considered in case of plants engaged exclusively in the manufacture of ice. Expressed in tons per 24 hr. this is equal to about 1.8 times the ice-melting capacity. This takes into account losses, chilling of the can water, etc. Rating in terms of ice-making capacity is therefore only definite when the temperatures of water and finished ice are stated.

The coefficient of performance is a more correct term for what is sometimes miscalled the efficiency of a refrigeration system. This quantity is the ratio of the refrigerating effect in B.t.u. per unit of time to the heat equivalent effect to the indicated work done by the compressor in the same time;

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that is, it is the useful effect divided by the power put in, and in this respect is superficially an efficiency. The term efficiency generally has reference to the relation of an energy after passing through a conversion to its value at the source. In the case of refrigeration machinery the compressor energy is not the source of the useful effect. In reality it is the condensing water that does the cooling, the compressor being merely an auxiliary in the process.

The ideal coefficient of performance is the maximum that could be obtained under a given set of operating conditions with no losses. It is fixed by the temperatures of condenser and refrigerator and is equal to

$$P_i = \frac{460 + T_R}{T_C - T_R}$$

where T_R is the temperature Fahrenheit of the refrigerator and T_C is the temperature of the condenser.

The efficiency of the plant may be expressed as the ratio of the actual to the ideal coefficient of performance.

The *economy* of compression plants may be expressed in pounds of refrigerating effect per pound of coal, on the assumption that a certain number of pounds of coal are consumed to make one indicated horsepower at the compressor.

For complete tests, enabling a study of all the heat transfers, temperature and time-quantity determination should be made of both ammonia and cooling water. Here only measurements as indicated by the above definitions will be considered.

The duration of the trial should be at least 12 hr., and preferably 24, to allow for the possible error at the start and end of the test.

Coming now to the measurement of the refrigerating effect, this, in B.t.u. per minute, equals

$$R = W(T - t)C$$

In which

W = weight of brine in pounds per minute;
T = temperature of brine at inlet in degrees Fahrenheit;
t = temperature of brine at outlet in degrees Fahrenheit;
C = specific heat of brine.

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For the determination of W the various methods of water measurement have been applied. Owing to the large quantity of brine circulated even in small plants, the method of direct weighing by tanks and scales is inconvenient. A modification of this consists in running the brine through two tanks, one above the other, the upper one having in its bottom a number of orifices through which the brine flows. The rate of flow varies with the height of the brine level in the upper tank. One of the orifices may be readily calibrated during the trial by collecting, weighing and timing its discharge at various heads. The total brine flowing through all the orifices may then be found by multiplication.

When meters calibrated in gallons or cubic feet are used, a separate determination of the weight of the brine per gallon or cubic foot is necessary. This may be made with a hydrometer, care being taken to test a sample withdrawn near the meter and at approximately the same temperature as it has in the pipe.

Concerning temperatures T and t, as the range is only between 5 and 10 deg., finely graduated thermometers must be used. For not more than 2 per cent of error the graduations must be about 2 per cent of the temperature range—that is, between one-fifth and one-tenth degree. The thermometers should be inserted in wells filled with mercury or oil and should be located in the inlet and outlet pipes as near as possible to the cooler. The protruding portions of the wells should be well insulated. During the trial it is well to interchange the thermometers as a check on their accuracy.

The specific heat of brine varies with its concentration, constituents and temperature. The last-named variation is comparatively small—about 0.06 per cent decrease of the specific heat for each degree decrease in temperature. As to the effect of the constituents, the presence in calcium chloride brine of manganese and sodium chloride in moderate quantities (say up to 20 per cent) does not affect the specific heat of the mixture materially. The effect of the concentration, however, is to lower the specific heat from unity (that of water) down

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to about 0.65 at a concentration corresponding to a specific gravity of 1.26.

The following formula for pure calcium chloride brine will give results for the specific heat of commercial calcium chloride brine to within 1 or 2 per cent of error, between the limits of −4 and 40 deg. F., and specific gravities between 1.10 and 1.26:

$$C = 1.833 - 0.93G - 0.0005(32 - Ta)$$

In this formula[1] G is the specific gravity, and Ta is the average temperature of the brine in degrees Fahrenheit at inlet and outlet.

Ice-melting capacity is readily calculable from the value of the refrigerating effect R in B.t.u. per min., by dividing by 200, the result being in tons per 24 hr.

To obtain the actual coefficient of performance, it is necessary to indicate the compressor. The process here is similar to that for steam-engine trials and hardly needs comment other than that a special steel-lined indicator must be used with as short pipe connections as possible to avoid materially increasing the clearance space. Having obtained the indicated horsepower of the compressor, the coefficient sought is

$$Pa = \frac{R}{42.4 \times i.hp.}$$

The ideal coefficient of performance may be found from the average thermometer readings at the brine and condenser water outlets, as previously defined. Or, if vapor pressures are used, the corresponding temperatures may be found from the ammonia tables. When the plant is equipped with pressure gages bearing thermometric scales, the ammonia tables may be dispensed with.

An an illustration of the difference that may exist between the values of the ideal coefficient as obtained by the two methods previously discussed, consider the following data:

Suction pressure, 28-lb. gage; from tables, temperature, 15 deg.; head pressure, 151-lb. gage; from tables, temperature,

[1] Deduced from the values given in Bureau of Standards *Bulletin* No. 135.

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84.5 deg.; outgoing brine temperature, 28.9 deg.; outgoing condenser-water temperature, 83.5 deg.

From these figures the ideal coefficient of the whole system is

$$\frac{460 + 28.9}{83.5 - 28.9} = 8.95$$

and for the system, exclusive of condenser and refrigerator transmission losses, is

$$\frac{460 + 15}{84.5 - 15} = 6.83$$

The measurement of the condenser water is simpler than that of the brine, the quantity being much less. Direct weighing or the usual forms of water meter may be employed, care being taken to select one appropriate to the conditions of flow, whether pulsating or steady. The number of gallons of cooling water per minute per ton of ice-melting capacity should be figured for purposes of comparison and also in order to add the cost of this item to the other costs to obtain the total.

The performance of auxiliaries may be either estimated or measured. When they and the compressor are steam driven, it is well to measure the total steam supplied to the plant by the boiler-feed method or by a steam meter installed so as to avoid inaccuracies due to pulsating flow. The steam consumption of the compressor and of the whole plant may then be stated in pounds of steam per ton of ice-melting.

Ratings are generally based on head and suction pressures of 185 and 15.7 gage respectively. Any departure from these conditions should be considered in making comparisons of test results.

TESTING ABSORPTION PLANTS

In some respects the results from and methods for the test of an absorption plant are identical to those applying to the compression system. The capacity in terms of ice-melting effect may be found in the same way, and the investigation of the condenser operation is the same. The actual coefficient of performance may not readily be obtained, since the energy

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imparted to the ammonia (barring the work of the ammonia pump, which is comparatively small) is in the form of heat instead of compressor work. But the ideal coefficient of performance may be determined as for a compression plant and may be useful for purposes of comparison of the two systems.

The principal economy result is the steam consumption expressed in pounds of steam per hour per ton of refrigerating effect. This may be determined *in toto* for the plant or itemized as live steam to the pump and steam to the generator, the latter being subdivided into live and exhaust steam according to whether the one or the other or a combination is used. Under favorable circumstances the generator will use 30 lb. of steam per hour per ton. The steam consumption of ammonia pumps is subject to wide variations due to their construction. Although the energy delivered is small compared with the heat transferred in the generator, a test of this pump should be included in economy trials because of the bearing of its performance on the steam consumption of the plant; and the pump test should preferably include a determination of the horsepower delivered.

Condition and Amount of Ammonia.—In addition to these results it is desirable to ascertain the condition of the aqua ammonia and, if convenient, the amount of anhydrous circulated per unit of time. The former is expressed in terms of its concentration; that is, the part of a pound of ammonia in 1 lb. of the aqua ammonia (more frequently expressed as a percentage). The concentration results show the working of the absorber and generator and enable calculations regarding the ammonia circulation.

The steam consumed by the generator is readily ascertained by piping the generator-trap discharge into a weighing barrel. The barrel, to start with, should be about half full of cold water to prevent evaporation of the condensate from the trap. Quick emptying should be provided for. Weights should be recorded at uniform time intervals (to show uniformity of operation), from which data the rate in pounds per hour may be had.

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Quality determinations of the steam are important, unless an efficient separator is installed to remove the moisture from the incoming steam, to the consideration of the results, as also are pressure readings.

If reduced live steam is used in addition to exhaust and it is sought to measure them separately, a steam meter may be applied to the live-steam line. The exhaust-steam quantity then equals the total trap condensate minus the meter reading. Measurement of the feed water, as for a boiler test, may also be resorted to, in cases where the plant may be isolated.

When conditions are not favorable to the separate determination of the steam quantities, the steam required to drive the pump may be estimated from the manufacturer's figures for its water rates and from the horsepower actually delivered. These remarks of course apply to steam pumps; where power pumps are installed, the energy may be considered in the list of costs.

Net Horsepower Important.—Now as to the energy delivered by the ammonia pump, it is preferable to find the net horsepower output rather than the indicated horsepower. From the data required for the net horsepower and from the number of pump displacements, the slip may be calculated as in ordinary pumping-engine trials.

The required formula for horsepower is

$$\text{Net horsepower} = \frac{144(P_1 - P_2)V}{33{,}000}$$

in which P_1 and P_2 are the gage pressures worked against in pounds per square inch (that is, of the generator and absorber, respectively) and V is the number of cubic feet of strong aqua circulated per minute.

The quantity of strong aqua V may be determined in several ways. A specially calibrated meter may be used on the discharge side of the pump; the aqua may be measured in calibrated tanks on the suction side; or it may be calculated from the amount of anhydrous circulating or from the quantity of weak aqua, the concentrations being known.

When a receiver is installed to supply the ammonia pump, it may be readily calibrated so that the volume contained,

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corresponding to any height of the liquid in the gage glass, is known. Now, by cutting off the flow from the absorber into this receiver for a short interval, the rate, in cubic feet per minute, at which the strong aqua is pumped out, may be ascertained. If desired, two receivers may be arranged in parallel, thereby enabling continuous measurement.

If either the anhydrous or weak aqua is measured instead of the strong aqua, the latter may be calculated from the following relations:

$$A_s = \frac{1 - C_w}{C_s - C_w} \times A$$

$$A_s = \frac{1 - C_w}{1 - C_s} \times A_w$$

in which A_s, A_w and A are the weights in pounds per minute of the strong and weak aqua and anhydrous respectively, and C_s and C_w are the concentrations of the strong and weak aqua respectively, in pounds of NH_3 per pound of solution.

It is to be noted that, since these relations are between weights, a knowledge of densities is necessary to convert them into volumes. For the aqua this involves a measurement of specific gravity; for the anhydrous the ammonia tables may be referred to. The accompanying chart in Fig. 99 gives values of the density of aqua corresponding to those of specific heat.

As an example of how these relations are to be applied, let us suppose that the concentrations of the strong and weak aqua are 38 and 26 per cent respectively (the method of determining concentration will be described later), and that the number of pounds of anhydrous circulated per minute is 20. Then

$$A_s = \frac{1 - 0.26}{0.38 - 0.26} \times 20 = 123 \text{ lb. per minute}$$

Referring to the chart, it is seen that 38 per cent aqua has a specific gravity of 0.873, if the aqua is at 60 deg. F., and a density of 54.4 lb. per cubic foot. If the temperature of the aqua at the pump is not much higher than this, then the cubic feet discharged per minute is $123 \div 54.4 = 2.26$, which is the value of V to be used in the formula for horsepower.

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The concentration of aqua ammonia is determinable from its specific gravity. Given a definite solution of ammonia in water, its specific gravity is fixed at any one temperature,

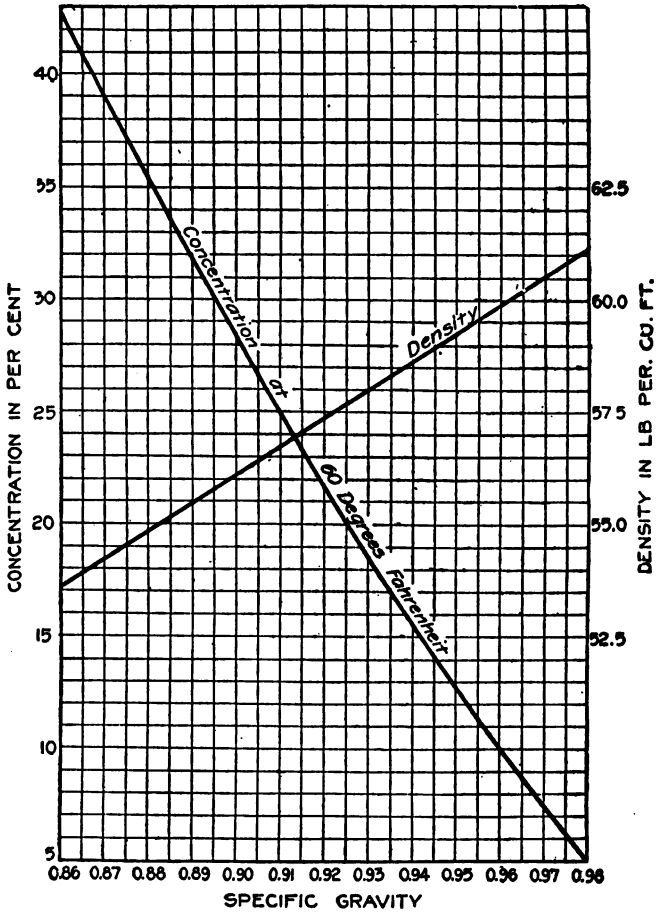


FIG. 99.—Density-specific gravity brine chart.

say 60 deg. At any higher temperature the solution occupies more volume, owing to expansion, and therefore has a lower specific gravity, although the concentration remains the same.

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Or to put it another way, for any value of the specific gravity the concentration depends on the temperature.

Because of the volatility of aqua ammonia at the high temperature, and reduced pressure prevailing when a sample is drawn, considerable care must be taken when testing for concentration. The following procedure is recommended. Outlets for samples of the aqua are arranged at points where the temperatures are comparatively low; that is, on the discharge side of the pump for the strong aqua and between the absorber and weak-aqua cooler for the weak. These outlets should be fitted with short lengths of rubber tubing. A glass graduate, reading preferably in cubic centimeters, is provided, together with a hydrometer or a sensitive scales. The graduate is about half filled with water for the first trial (for the second, a somewhat different amount may be selected, depending upon the outcome of the first) and then placed in a bucket of cold brine until it and its contents are chilled to about 32 deg. After running a little aqua through the sampling tube, it is directed below the surface of the water in the graduate until the mixture of cold water and incoming aqua attains a temperature of about 60 deg. as shown by a thermometer. The specific gravity of the mixture should now be taken quickly with a hydrometer or by weighing the contents of the graduate and then by calculation. Provided the temperature is within 5 to 10 deg. of 60 deg., the concentration of the mixture may now be found on the chart against the determined value of its specific gravity. It is the concentration of the sample that is sought, however, and this may be calculated from the following relation:

Concentration of sample =

$$\text{concentration of mixture} \div \left(1 - \frac{R}{S_m}\right)$$

in which R is the ratio of the volume of cold water, before mixing with the sample, to the volume of the mixture, and S_m is let represent the specific gravity of the mixture.

If scales instead of hydrometer are used to obtain the weights of cold water and mixture, then the concentration of the sample is more readily figured from the relation.

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Concentration of sample =

$$\text{concentration of mixture} \times \frac{\text{weight of mixture}}{\text{weight of sample}}$$

If the temperature of aqua ammonia is higher than 60 deg., its specific gravity will be lower by between 0.001 and 0.005 for each 10 deg., depending upon the concentration, the stronger solutions having a higher coefficient of expansion and there fore needing greater correction than the weaker. Under these circumstances the specific gravity S_{60} of a given solution at 60 deg. can be figured from its specific gravity S_t at t deg., the following formula, which gives moderately accurate results, being used:

$$S_{60} = S_t + 0.003(t - 60)(1 - S_t)$$

For example, if the specific gravity is found to be 0.865 at 75 deg., then at 60 deg. it equals

$$S_{60} = 0.865 + 0.003(75 - 60)\ (.1 - 0.865) = 0.871$$

The value 0.871 is then to be referred to the chart Fig. 99 for the corresponding concentration.

Concentration of Aqua Ammonia at 60 deg. F., and Density Variation.—When the reverse calculation is sought (for the density determination of aqua at high temperatures) this formula may be expressed:

$$S_t = \frac{S_{60} - 0.003(t - 60)}{1 - 0.003(t - 60)}$$

Turning now to the measurement of the quantity of weak aqua and anhydrous circulated per minute, calculations may be based on the relations previously given, transposed as follows:

$$A_w = \frac{1 - C_s}{1 - C_w} \times A_s$$

$$A = A_s - A_w$$

For direct measurements of these quantities, much the same methods may be employed as for the strong aqua.

HEAT BALANCE OF AN ABSORPTION PLANT

The overall heat transfers in an absorption refrigeration plant may be summarized very briefly as follows: Energy is

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added to the system, first by the steam supplied to the generator, second by the work of the ammonia pump, and third by the heat added to the brine during the refrigerating process. Heat is removed from the system by circulating water, first in the ammonia condenser, second in the absorber, and lastly, in the weak-aqua cooler and the rectifier. The energy added equals the heat removed, as itemized, plus or minus radiation.

A study of this overall heat balance is useful, but before making comparisons of grand totals, it is preferable to analyze the performance of separate units. Of these it is the purpose here to deal with the generator and the absorber.

To review the action of the generator, strong aqua ammonia enters at a comparatively low temperature, to which heat is added by steam pipes. Some of the ammonia is thereby boiled out of the water holding it in solution. There result from the process a quantity of superheated anhydrous ammonia, mixed with a small percentage of steam, and a larger quantity of weak aqua, comparatively hot, which is to be returned to the absorber. The heat in the steam supplied goes to vaporize and superheat ammonia from the form of strong aqua and to raise the temperature of the entering aqua to a higher value upon leaving. Furthermore, a certain quantity of heat is required to break the bond between the liquid ammonia and the water holding it in solution, before vaporization can take place. This quantity is additional to that required to vaporize liquid anhydrous ammonia and is referred to as the "heat of solution." Experimental values of it have been made showing that it is independent of pressure and temperature and varies only with the concentration of the solution, being about 347 B.t.u. per pound for very weak solutions (approximately to zero percentage of ammonia) and diminishing as the concentration is increased to a zero value when the concentration is 60 per cent.

It is convenient to base the heat calculations of the ammonia upon a single pound of the NH_3 going through a complete cycle of temperature and state changes. From these results hourly quantities may be figured by multiplying by the number of pounds of anhydrous circulated per hour. Thus, let A

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represent the latter quantity and W the number of pounds of weak aqua circulated per pound of anhydrous vaporized; then the heat transferred in the generator per hour = $A \times$ ($W \times$ heat added to 1 lb. of the aqua + heat of solution in B.t.u. per pound of NH_3 + difference in "heat content" of NH_3 per pound, from tables). The quantity in the parenthesis is the heat transferred for the circulation of 1 lb. of the anhydrous.

The product, representing B.t.u. per hour, must equal the heat given up by the steam supplied to the generator, if the losses due to radiation and steam in the superheated ammonia are ignored.

Measurements Necessary in Heat Balance.—The measurements necessary to make for the determination of this heat balance are in part the same as those for an economy trial. In addition it is necessary to determine temperatures of the ingoing and outgoing aqua near the generator, the temperature of the ammonia vapor before it enters the rectifier and the head pressure—readings which should be taken sufficiently to get fair averages. The vapor data are used to determine the "heat content" from the tables or Mollier diagram for ammonia. For the steam-heat quantities, pressure and quality must be ascertained as well as the temperature of the condensate at the generator trap.

To determine the number of pounds of weak aqua circulated per pound of anhydrous (that is, W), it is necessary to measure the concentrations of the weak and strong aqua (referred to as C_w and C_s, respectively) as described in the article previously mentioned. The desired quantity may then be calculated from the formula,

$$W = \frac{1 - C_s}{C_s - C_w}$$

A simpler procedure may be adopted through the use of the accompanying curves. Referring to Fig. 100, the lines slanting diagonally upward from left to right show values of the number of pounds of weak aqua per pound of anhydrous corresponding to various combinations of concentrations. To use these curves, assume, for example, that the concentra-

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tion of the strong aqua is 0.34, and of the weak, 0.24. On the chart the lines representing these values intersect between the curves marked 6.5 and 7, at about one-fifth of the distance across, so the value of W is 6.6.

This chart also gives the "heat of solution," which is plotted from the experimental values of H. Mollier. As previously remarked, this depends only upon the concentration. During

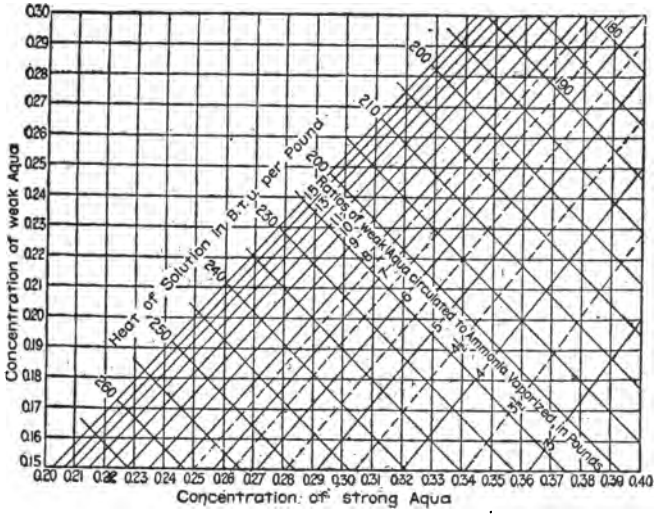


FIG. 100.—Heat of aqua ammonia solution.

the driving-off process in the generator, the concentration varies from that of the strong aqua to that of the weak. It is the average concentration then, that determines the heat of solution. This condition is met in the chart, as before, by the intersection of the lines representing the concentrations; the location of this point with relation to the "heat lines" (slanting downward from left to right) gives the heat of solution. For example, using the concentrations 0.34 and 0.24, the point of intersection is about four-ffiths of the distance across from the 205 to the 210 B.t.u. line. Therefore the heat of solution is 209 B.t.u.

We now have means of finding the number of pounds of weak aqua per pound of ammonia vaporized and the heat of

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solution per pound. Referring to the heat equation for the generator, already given, it will be noticed that there are still to be found the heat added to 1 lb. of the weak aqua and the difference in "heat content" of the NH_3 before and after vaporization. The latter is readily found from the ammonia tables or Mollier diagram, being the difference between the total heat of the NH_3 vapor at the head pressure and outgoing temperature and the heat of the liquid NH_3 at the temperature of the incoming strong aqua.

Heat Capacity Ammonia and Water Mixture.—Considering, now, the heat added to 1 lb. of the aqua passing through the generator, it is to be remarked that there are no tabulated values of the heat of the liquid of mixtures of ammonia and water, or of the specific heats of such mixtures. It will, however, be sufficiently accurate to assume that the heat capacity of a combination of ammonia and water in definite proportion is the sum of the heat capacities of the constituents, according to their proportion. Thus, if we call the heat of the liquid ammonia q_a and of the water, q_w (as tabulated in the ammonia and steam tables, respectively), then for a mixture of C lb. of ammonia and $1 - C$ lb. of water, the heat of the liquid (aqua ammonia) $= qaC + q_w(1 - C)$. Values of this quantity at various temperatures and concentrations are shown graphically in Fig. 101, the highest curve of which gives (vertically) the heat of the liquid of pure ammonia, the lowest curve, of unmixed water, and the intermediate curves, of mixtures of the two denoted by their concentrations. The heat of the liquid of aqua at concentrations intermediate between those shown may be obtained with sufficient accuracy by interpolation.

Coming now to detailed calculations, let us assume the following data:

Concentration, weak aqua, $C_w = 0.24$.
Concentration, strong aqua, $C_s = 0.34$.
Head pressure = 100.3-lb., gage.
Temperature of outgoing NH_3 = 150 deg. F.
Temperature, weak aqua = 180 deg. F.
Temperature, strong aqua = 130 deg. F.

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From Fig. 100 is obtained the value 6.6 as the number of pounds of weak aqua leaving the generator per pound of ammonia vaporized. From Fig. 101 the heat of the liquid of 1 lb. of 24 per cent aqua is found to be 158 B.t.u. at 180 deg. and 102 B.t.u. at 130 deg. Consequently, the heat added to the aqua per pound of ammonia vaporized is 6.6 × (158 —

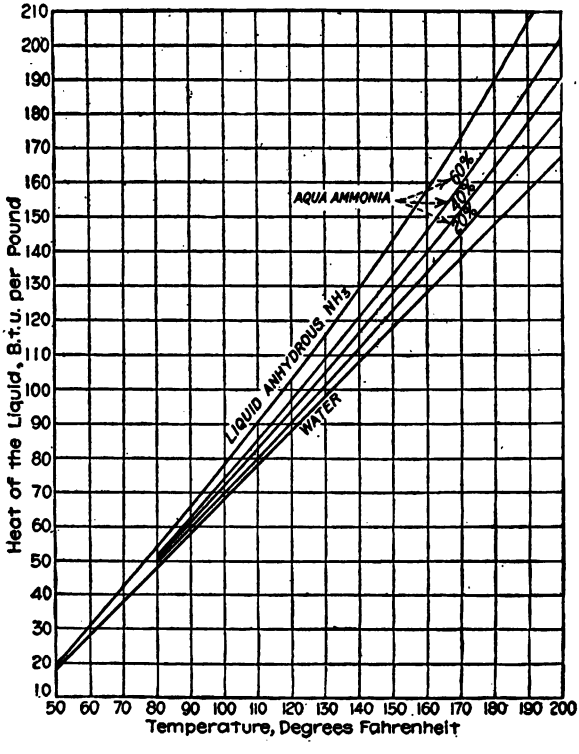


FIG. 101.—Heat of liquid for water, aqua ammonia and anhydrous ammonia.

102) = 370 B.t.u. From Fig. 100, of the givencon centrations heat of solution = 209 B.t.u. From the ammonia tables, the total heat of ammonia at 100.3-lb. gage and 150 deg. is 609 B.t.u., and the heat of liquid ammonia at 130 deg. (its incoming temperature) is 115 B.t.u. (also obtainable from Fig. 2). The heat added to the NH_3 to raise it from the liquid to the

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superheated condition is consequently 609 − 115 = 494 B.t.u. Adding these three heat quantities (370 + 209 + 494 = 1,073 B.t.u.), we have the heat added in the generator to effect the complete circulation through the plant of 1 lb. of anhydrous ammonia. As before mentioned, this ignores the loss through vaporization of water with the ammonia, afterward removed by the rectifier, but this loss should be small.

The steam-heat quantities involve the determination of the weight of steam used for a corresponding ammonia vaporization and may be made as for an economy trial. It should be noted that

$$\text{Pounds of steam per hour} \frac{\text{heat added to } NH_3 \text{ and aqua per hour}}{\text{heat removed from 1 lb. of steam}}$$

The heat transfers in the absorber are essentially the same as in the generator, the only difference being that they proceed from the ammonia media into circulating water, instead of from steam into ammonia media, and heat is given up, instead of taken in, by the NH_3 and aqua. The aqua is pumped from the absorber at a temperature lower than that at which it entered, whereas in the generator the reverse was the case. The heat of solution is *released* in the absorber and must be removed by the circulating water, also a reverse process. The anhydrous vapor, coming from the refrigerator at comparatively low pressure, is taken into solution by the weak aqua, consequently giving up (in addition to the heat of solution) an amount of heat equal to the total heat of the incoming anhydrous minus the heat of the liquid NH_3 at the temperature of the outgoing (strong) aqua. It will thus be seen that the measurements and calculations for the determination of the heat given up by the absorber per pound of anhydrous ammonia are exactly the same as for the generator.

The heat thus calculated, expressed in B.t.u. per unit of time, equals the heat removed by the circulating water from the absorber at the same time, minus or plus radiation. To obtain the heat removed, it is necessary to measure the circulating water for a definite time and also its "in" and "out" temperatures.

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PLANT LOGS

Each refrigeration plant should keep an accurate record of all plant expense. A monthly log is shown below, Fig. 102. To obtain the information required in this sheet a daily log

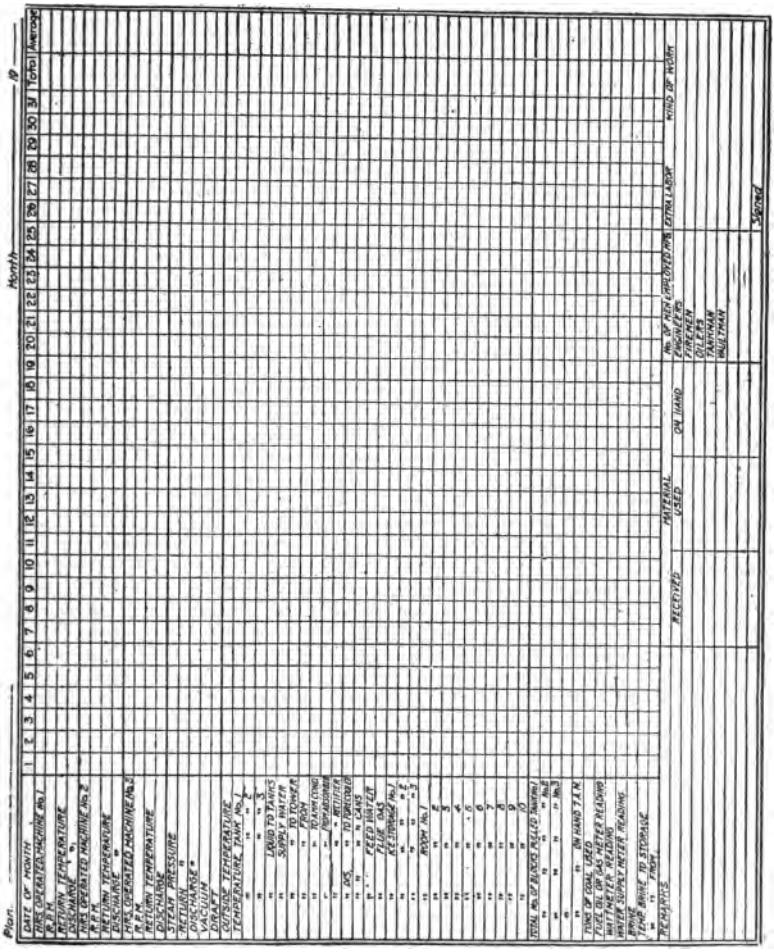


Fig. 102.—Monthly plant report.

such as Fig. 103 is essential. This log sheet by providing several readings of temperatures, fuel, oil, etc. each day allows the engineer to have a deeper insight into the plant's condition.

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Plant _____ Date _____ 19 _____

TIME	STEAM PRESSURE	CONDENSER PRESSURE (MACHINE No. 1 2 3)	TEMPERATURE OF DISCHARGE GAS (MACHINE No. 1 2 3)	RETURN PRESSURE (MACHINE No. 1 2 3)	TEMPERATURE RETURN VAPOR (MACHINE No. 1 2 3)	R. P. M (MACHINE No. 1 2 3)	WATER TEMPERATURE: TO HEATER	TO BOILER	TO COOLING TOWER	FROM COOLING TOWER	TO CONDENSER	FROM CONDENSER	SUPPLY	FROM ABSORBER	FROM RECTIFIER	TO PRECOOLER	TO CANS	BRINE TEMP. TO STORAGE	FROM STORAGE	TEMPERATURE FREEZING TANK (TANK No. 1 2 3)	TEMPERATURE ROOMS (ROOM No. 1 2 3 4 5 6)	TEMPERATURE ICE STORAGE	VACUUM GAGE	DRAFT GAGE	OUTSIDE TEMPERATURE	FUEL USED	CO_2
7 A.M.																										
10 A.M.																										
1 P.M.																										
4 P.M.																										
7 P.M.																										
10 P.M.																										
1 A.M.																										
4 A.M.																										
A.V.																										

MACHINE NO	START	STOP	START	STOP	TOTAL
1					
2					
3					

TOTAL No. BLOCKS PULLED, DAY
" " " " NIGHT
" " " IN ICE HOUSE
" " " IN DUMP ROOM
TOTAL TONS PULLED
TOTAL TONS ON HAND, 7 A.M.
QUALITY OF ICE

METER READINGS 7 A.M.

OIL
GAS
WATT
WATER
BRINE
TONS OF COAL USED
CARS SHIPPED

ENGINEERS
FIREMEN
TANKMAN
VAULTMEN
EXTRA LABOR
CHIEF ENGINEER

MATERIAL RECEIVED

MATERIAL USED

REMARKS (SHUT DOWNS, ACCIDENTS ETC)

Signed _____

FIG. 103.—Daily plant log.

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APPENDIX

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TABLE 15.—STANDARD DIMENSIONS EXTRA HEAVY PIPE

DIAMETER			Thickness Inches	CIRCUMFERENCE		LENGTH OF PIPE PER SQUARE FT.		TRANSVERSE AREAS			Nominal Weight Pounds Per Foot	Number of Threads Per Inch
Nominal Inches	External Inches	Internal Inches		External Inches	Internal Inches	External Surface Sq. Inches	Internal Surface Sq. Inches	External Square Inches	Internal Square Inches	Metal Square Inches		
⅛	.405	.205	.100	1.272	.644	9.44	18.63	.129	.033	.096	.29	27
¼	.540	.294	.123	1.696	.924	7.07	12.99	.229	.068	.161	.54	18
⅜	.675	.421	.127	2.121	1.323	5.66	9.07	.358	.139	.219	.74	18
½	.840	.542	.149	2.639	1.703	4.55	7.05	.554	.231	.323	1.09	14
¾	1.050	.736	.157	3.299	2.312	3.64	5.11	.866	.425	.441	1.39	14
1	1.315	.951	.182	4.131	2.988	2.90	4.02	1 358	.710	.648	2.17	11½
1¼	1.660	1.272	.194	5.215	3.996	2.30	3.00	2.164	1.271	.893	3.00	11½
1½	1.900	1.494	.203	5.969	4.694	2.01	2.56	2.835	1.753	1.082	3.63	11½
2	2.375	1.933	.221	7.461	6.073	1.61	1.97	4.430	2.935	1.495	5.02	11½
2½	2.875	2.315	.280	9.032	7.273	1.33	1.65	6.492	4.209	2.283	7.67	8
3	3.500	2.892	.304	10.996	9.086	1.09	1.33	9.621	6.569	3.052	10.25	8
3½	4.000	3.358	.321	12.566	10.549	.955	1.14	12.566	8.856	3.710	12.47	8
4	4.500	3.818	.341	14.137	11.995	.849	1.00	15.904	11.449	4.455	14.97	8
4½	5.000	4.280	.360	15.708	13.446	.764	.893	19.635	14.387	5.248	18.22	8
5	5.563	4.813	.375	17.477	15.120	.687	.793	24.306	18.193	6.113	20.54	8
6	6.625	5.751	.437	20.813	18.067	.577	.664	34.472	25.976	8.496	28.58	8
7	7.625	6.625	.500	23.955	20.813	.501	.598	45.664	34.472	11.192	37.67	8
8	8.625	7.625	.500	27.096	23.955	.443	.502	58.426	45.664	12.762	43.00	8
9	9.625	8.625	.500	30.238	27.096	.397	.443	72.760	58.426	14.334	48.25	8
10	10.750	9.750	.500	33.772	30.631	.355	.399	90.763	74.662	16.101	54.25	8
12	12.750	11.750	.500	40.055	36.914	.299	.325	127.68	108.43	19.25	65.00	8

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TABLE 16.—COMPARATIVE CAPACITIES OF PIPES OF STANDARD SIZES, SHOWING THE NUMBER OF TIMES THE AREA OF ONE PIPE IS CONTAINED IN THAT OF A LARGER SIZE

	⅛	¼	⅜	½	¾	1	1¼	1½	2	2½	3	3½	4	5	6	7	8	9	10
⅛	1																		
¼	1 9	1																	
⅜	3 3	1 8	1																
½	5 3	2 9	1 5	1															
¾	9 3	5 2	2 7	1 7	1														
1	14 7	7 7	4 1	2 6	1 6	1													
1¼	26 2	14 3	7 8	4 9	2 8	1 6	1												
1½	35 6	19 5	10 6	6 6	3 8	2 4	1 3	1											
2	58 6	32 2	17 5	11 0	6 9	3 9	2 2	1 6	1										
2½	83 6	44 6	25 0	15 7	9 0	5 5	3 2	2 3	1 4	1									
3	129 1	71 0	38 5	24 2	13 8	8 5	5 0	3 6	2 2	1 5	1								
3½	172 8	95 0	51 6	32 4	18 5	11 4	6 6	4 8	2 9	2 0	1 3	1							
4	222 5	122 3	66 4	41 7	23 8	14 7	8 5	6 2	3 8	2 6	1 7	1 3	1						
5	349 4	192 0	104 3	65 6	37 4	23 1	13 3	9 8	5 9	4 1	2 7	2 0	1 5	1					
6	505 0	277 5	150 7	94 7	54 1	33 4	19 3	14 1	8 6	6 0	3 9	2 9	2 2	1 4	1				
7	677 2	372 1	202 2	127 0	71 3	44 9	25 9	19 0	11 5	8 1	5 2	3 9	3 0	1 9	1 3	1			
8	878 3	480 7	261 2	164 1	93 8	58 0	34 5	24 5	14 9	10 4	6 7	5 0	3 9	2 5	1 7	1 3	1		
9	1112 4	611 2	332 1	208 7	179 3	73 4	42 7	31 2	19 0	13 3	8 7	6 4	5 0	3 1	2 2	1 6	1 2	1	
10	1378. 3	757 3	411 5	258 6	147 8	91 3	52 7	38 6	23 5	16 5	10 6	7 9	6 2	3 9	2 7	2 0	1 5	1 2	1

TABLE 17.—WROUGHT IRON WELDED STEAM, GAS AND WATER PIPE
Table of Standard Dimensions

Diameter			Thickness Inches	Circumference		Transverse Areas			Length of Pipe per Sq. Foot of		Length of Pipe Cont'g One Cu. Foot	Weight per Ft. of Leth. Pounds	No. of Threads per Inch of Screw	Contents in Gal. per Foot of Leth.	Weight of Water per Foot of Leth. Pounds
Nom. Inter'l Is.	Actual External Inches	Actual Internal Inches		External Inches	Internal Inches	External Sq. Inch	Internal Sq. Inch	Metal Sq. Inch	External Surface Feet	Internal Surface Feet					
⅛	.405	.27	.068	1.272	.848	.129	.0573	.0717	9.44	14.15	2513.	.241	27	.0006	.005
¼	.54	.364	.088	1.696	1.144	.229	.1041	.1249	7.075	10.49	1383.3	.42	18	.0026	.021
⅜	.675	.494	.091	2.121	1.552	.358	.1917	.1863	5.657	7.73	751.2	.539	18	.0057	.047
½	.84	.623	.109	2.639	1.957	.554	.3048	.2492	4.547	6.13	472.4	.837	14	.0102	.085
¾	1.05	.824	.113	3.299	2.589	.866	.5333	.3327	3.637	4.635	270	1.115	14	.0230	.190
1	1.315	1.048	.134	4.131	3.292	1.358	.8626	.4954	2.904	3.645	166.9	1.668	11½	.0408	.340
1¼	1.66	1.38	.14	5.215	4.335	2.164	1.496	.668	2.301	2.768	96.25	2.244	11½	.0638	.527
1½	1.9	1.611	.145	5.969	5.061	2.835	2.038	.797	2.01	2.371	70.66	2.678	11½	.0918	.760
2	2.375	2.067	.154	7.461	6.494	4.43	3.356	1.074	1.608	1.848	42.91	3.609	11½	.1632	1.354
2½	2.875	2.468	.204	9.032	7.753	6.492	4.784	1.708	1.328	1.547	30.1	5.739	8	.2550	2.116
3	3.5	3.067	.217	10.996	9.636	9.621	7.388	2.243	1.091	1.245	19.5	7.536	8	.3673	3.049
3½	4.	3.548	.226	12.566	11.146	12.566	9.887	2.679	.955	1.077	14.57	9.001	8	.4998	4.155
4	4.5	4.026	.237	14.137	12.648	15.904	12.73	3.174	.849	.949	11.31	10.665	8	.6528	5.405
4½	5.	4.508	.246	15.708	14.162	19.635	15.961	3.674	.764	.848	9.02	12.34	8	.8263	6.851
5	5.563	5.045	.259	17.477	15.849	24.306	19.99	4.316	.687	.757	7.2	14.502	8	1.020	8.500
6	6.625	6.065	.28	20.813	19.054	34.472	28.888	5.584	.577	.63	4.98	18.762	8	1.469	12.312
7	7.625	7.023	.301	23.955	22.063	45.664	38.738	6.926	.501	.544	3.72	22.371	8	1.999	16.662
8	8.625	7.982	.322	27.096	25.076	58.426	50.04	8.386	.443	.478	2.88	26.177	8	2.611	21.750
9	9.625	8.937	.344	30.238	28.076	72.76	62.73	10.03	.397	.427	2.29	33.701	8	3.309	27.589
10	10.75	10.019	.366	33.772	31.477	90.763	78.839	11.924	.355	.382	1.82	40.063	8	4.081	34.020
11	12.	11.25	.375	37.699	35.343	113.098	99.402	13.696	.318	.339	1.456	45.95	8	5.163	43.020
12	12.75	12.	.375	40.055	37.7	127.677	113.098	14.579	.299	.319	1.27	48.985	8	5.875	48.930

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TABLE 18.—PROPERTIES OF AMMONIA LIQUOR

(Starr's Table)

Showing the Relation between Pressure and Temperature for Solutions of Ammonia in Water of Different Strengths

Line	Per Cent of NH_3 by Weight and Degree Beaumé	0	5	10	15	20	25	30	35	40	45	50	55	60	Line
1	1	206.3	223.6	234.9	247.4	256.2	263.8	270.4	277.1	282.8	288.1	292.9	297.5	301.9	1
2	1.84-11°B	201.4	219.3	231.6	243.0	251.7	259.4	266.4	272.7	278.4	283.7	288.5	293.1	297.5	2
3	2	201.1	218.5	230.8	242.1	250.9	258.6	265.5	271.9	277.6	282.8	287.7	292.2	296.7	3
4	3	195.8	213.2	225.5	236.6	245.6	253.3	260.2	266.8	272.3	277.5	282.4	286.9	291.4	4
5	3.80-12°B	191.5	208.8	221.0	232.3	241.0	248.7	255.7	262.0	267.7	272.9	277.8	282.4	286.8	5
6	4	190.5	207.7	220.0	231.2	240.0	247.6	254.7	260.9	266.7	271.8	276.7	281.4	285.7	6
7	5	185.2	202.4	214.6	225.8	234.6	242.2	249.3	255.6	261.4	266.5	271.4	276.1	280.4	7
8	5.30-13°B	183.5	200.7	212.8	224.1	232.8	240.5	247.5	253.8	259.6	264.8	270.2	274.8	279.2	8
9	6	180.0	197.1	209.2	220.5	229.2	237.0	243.9	250.2	256.1	261.2	266.7	271.2	275.6	9
10	6.80-14°B	175.8	193.0	205.0	216.2	224.9	232.6	239.6	246.0	251.8	257.0	262.1	266.7	271.1	10
11	7	174.9	192.1	204.0	215.3	223.9	231.7	238.6	245.1	250.8	256.1	261.1	265.8	270.1	11
12	8	170.0	187.2	199.1	210.3	218.9	226.9	233.7	240.1	245.9	251.2	256.2	260.8	265.2	12
13	8.22-15°B	168.8	185.8	197.8	209.0	217.7	225.4	232.4	238.6	244.2	249.3	254.1	258.7	263.1	13
14	9	165.4	182.5	194.5	205.3	214.3	222.0	229.0	235.2	240.8	245.9	250.7	255.3	259.7	14
15	10-16°B	160.8	177.7	189.6	200.6	209.2	216.9	223.9	230.1	235.5	240.6	245.4	250.0	254.4	15
16	11	156.4	173.2	185.1	196.1	204.7	212.4	219.4	225.6	231.0	236.1	240.9	244.5	249.9	16
17	12	151.9	168.9	180.6	191.9	200.6	208.3	214.8	221.0	226.4	231.5	236.4	240.0	245.4	17
18	12.17-17°B	151.0	168.0	179.9	191.0	199.6	207.3	213.6	219.6	225.0	230.3	234.4	239.0	243.4	18
19	13	147.5	164.4	176.4	187.4	196.1	203.7	210.1	216.1	221.4	226.8	230.8	235.5	239.9	19
20	13.88-18°B	143.7	160.5	172.3	183.4	192.0	199.7	206.0	212.1	217.6	222.7	227.2	231.8	236.2	20
21	14	143.2	160.0	171.8	182.9	191.5	199.2	205 5	211.6	217.1	222.2	226.7	231.3	235.7	21
22	15	139.0	155.8	167.6	178.7	187.3	195.0	201.3	207.4	212.9	218.0	222.5	227.1	231.5	22
23	16	134.8	151.6	163.4	174.5	183.1	190.8	197.1	203.2	208.7	213.8	218.3	222.9	227.3	23
24	16.22-19°B	133.8	150.6	162.3	173.3	181.4	189.5	196.0	201.8	207.2	212.3	217.1	221.7	226.1	24
25	17	130.6	147.4	159.1	170.1	178.2	186.3	192.8	198.6	203.9	209.1	213.9	218.5	222.9	25
26	18.03-20°B	126.2	142.9	154.6	165.6	174.2	181.9	188.9	195.1	200.7	205.7	209.5	214.1	218.5	26
27	19	122.3	138.9	150.7	161.6	170.3	177.9	185.0	191.1	196.8	201.7	205.6	210.1	214.6	27
28	19.87-21°B	119.4	135.9	147.6	158.6	167.2	174.9	181.5	187.2	192.5	197.5	202.3	206.9	211.3	28
29	20	118.9	135.5	147.1	158.2	166.7	174.4	181.1	186.7	192.1	197.0	201.9	206.4	210.8	29
30	21	115.2	131.8	143.4	154.5	163.0	170.7	177.4	183.0	188.4	193.3	198.2	202.7	207.1	30
31	21.75-22°B	112.9	129.4	141.0	151.9	160.5	168.2	174.6	180.1	185.3	190.3	195.1	199.7	204.1	31
32	22	112.0	128.5	140.1	151.0	159.6	167.3	173.7	179.2	184.4	189.4	194.2	198.8	203.2	32
33	23.03-23°B	108.0	124.5	136.1	147.0	155.6	163.3	170.0	175.4	180.2	185.2	190.0	194.6	199.0	33
34	24	104.8	121.3	132.9	143.8	152.4	160.1	166.8	172.2	177.0	182.0	186.8	191.4	195.8	34
35	24.99-24°B	101.5	117.8	129.3	140.1	148.6	156.3	163.0	168.4	173.6	178.6	183.2	187.8	192.2	35
36	26	96.3	114.6	126.2	136.9	145.5	153.1	159.8	165.3	170.4	175.5	179.9	184.7	189.1	36
37	27	95.1	111.4	123.1	133.7	142.3	150.0	156.6	162.1	167.2	172.4	176.7	181.5	185.9	37
38	27.66-25°B	93.0	109.4	121.0	131.7	140.1	147.9	154.5	159.9	165.1	170.3	174.4	178.9	183.3	38
39	28	92.0	108.3	120.0	130.6	139.1	146.8	153.4	158.9	164.0	169.3	173.3	177.9	183.2	39
40	29	88.9	105.2	117.0	127.5	136.0	143.8	150.3	155.8	161.0	166.2	170.2	174.8	180.2	40
41	29.60-26°B	87.0	103.3	114.7	125.4	133.9	141.6	148.2	153.8	159.0	164.3	168.1	172.7	178.1	41
42	30	85.8	102.1	113.5	124.2	132.7	140.4	147.0	152.6	157.8	163.1	166.9	171.6	176.9	42
43	31.05-27°B	82.6	98.8	110.2	120.9	129.4	137.1	143.5	149.2	154.5	159.8	163.6	168.3	173.5	43
44	32	80.1	96.2	107.6	118.3	126.8	134.5	140.9	146.6	151.9	157.2	161.0	165.7	170.9	44
45	33	77.4	93.5	104.9	115.6	124.1	131.8	138.7	143.5	149.2	154.5	158.3	163.0	168.2	45
46	33.25-28°B	76.5	92.6	103.9	114.6	123.1	130.8	137.8	143.0	148.3	153.6	157.4	162.1	167.3	46
47	34	74.6	90.7	102.0	112.7	121.2	128.9	135.9	141.1	146.4	151.7	155.5	160.2	165.4	47
48	35	72.0	88.1	99.4	110.1	118.6	126.3	133.3	138.5	143.8	149.1	152.9	157.6	162.8	48

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TABLE 18.—PROPERTIES OF AMMONIA LIQUOR (*Continued*)

(Starr's Table)

Showing the Relation between Pressure and Temperature for Solutions of Ammonia in Water of Different Strengths

Line	Per Cent of NH_3 by Weight and Degree Beaumé	65	70	75	80	85	90	95	100	105	110	115	120	125	Line
1	1	306.3	310.4	314.4	318.2	321.8	325.2	328.5	331.7	334.8	337.8	340.7	343.5	346.2	1
2	1.84-11°B	301.8	306.0	310.0	313.8	317.4	320.8	324.1	327.3	330.4	333.4	336.3	339.1	341.8	2
3	2	300.9	305.2	309.2	312.9	316.6	320.0	323.2	326.5	329.6	332.6	335.4	338.2	341.0	3
4	3	295.6	300.0	303.9	307.6	311.3	314.7	317.9	321.2	324.3	327.3	330.1	332.9	335.7	4
5	3.80-12°B	291.1	295.3	299.3	303.1	306.7	310.1	313.4	316.6	319.7	322.7	325.6	328.4	331.1	5
6	4	290.1	294.2	298.3	302.1	305.6	309.1	312.4	315.5	318.7	321.6	324.5	327.4	330.0	6
7	5	284.8	288.9	293.0	296.3	300.3	303.8	307.1	310.2	313.4	316.3	319.2	322.1	324.7	7
8	5.30-13°B	283.5	287.7	291.7	295.5	299.1	302.5	305.8	309.0	312.1	315.1	318.0	320.8	323.3	8
9	6	280.0	284.1	288.2	291.9	295.5	299.0	302.2	305.5	308.5	311.6	314.4	317.3	319.7	9
10	6.80-14°B	275.4	279.6	283.6	287.4	291.0	294.4	297.7	300.9	304.0	307.0	309.9	312.7	315.2	10
11	7	274.5	278.6	282.7	286.4	290.1	293.5	296.7	300.0	303.0	306.1	308.9	311.8	314.2	11
12	8	269.6	273.7	281.7	281.5	285.2	288.6	291.7	295.1	298.1	301.2	303.9	306.9	309.3	12
13	8.22-15°B	267.4	271.6	275.6	279.4	283.0	286.4	289.7	292.9	296.0	299.0	301.9	304.7	307.2	13
14	9	264.0	268.2	272.2	276.0	279.6	283.0	286.3	289.6	292.6	295.6	308.5	301.3	303.8	14
15	10-16°B	258.7	262.9	266.9	270.7	274.3	277.7	281.0	284.2	287.3	290.3	293.2	296.0	298.5	15
16	11	254.2	258.4	262.4	266.2	268.8	273.2	276.5	279.7	282.8	285.8	288.7	291.5	294.0	16
17	12	249.8	253.9	257.9	261.7	264.3	268.7	272.0	275.2	278.3	281.3	289.2	287.0	289.5	17
18	12.17-17°B	247.7	251.9	255.4	259.7	263.3	266.7	270.0	273.2	276.3	279.3	282.2	285.0	287.5	18
19	13	244.2	248.4	251.8	256.2	259.8	263.1	266.5	269.6	272.8	275.7	278.6	281.5	284.0	19
20	13.88-18°B	240.5	244.0	248.7	252.5	256.1	259.5	262.8	266.0	269.1	272.1	275.0	277.8	280.3	20
21	14	240.0	243.5	248.2	252.0	255.6	259.0	262.3	265.5	268.6	271.6	274.5	277.3	280.0	21
22	15	235.8	239.3	244.0	247.8	251.4	254.8	258.1	261.3	264.4	267.4	270.3	273.1	275.8	22
23	16	231.6	235.1	239.8	243.6	247.2	250.6	253.7	257.1	260.2	263.2	266.1	268.9	271.6	23
24	16.22-19°B	230.4	234.6	238.6	242.4	246.0	249.4	252.7	255.9	259.0	262.0	264.9	267.7	270.2	24
25	17	227.2	231.4	235.4	239.2	242.8	246.2	249.5	252.7	255.8	258.8	261.7	264.5	267.0	25
26	18.03-20°B	222.8	227.0	231.0	234.8	238.4	241.8	245.1	248.3	251.4	254.4	257.3	260.1	262.6	26
27	19	218.8	223.1	227.0	230.9	234.4	237.9	241.1	244.4	247.4	250.5	253.4	256.1	258.7	27
28	19.87-21°B	215.6	219.8	223.8	227.6	231.2	234.6	237.9	241.1	244.2	247.2	250.1	252.9	255.4	28
29	20	215.2	219.3	223.4	227.1	230.7	234.1	237.4	240.7	243.8	246.7	249.6	252.4	255.0	29
30	21	211.5	215.6	219.7	223.3	227.0	230.4	233.7	237.0	240.1	243.0	245.9	248.7	231.3	30
31	21.75-22°B	208.4	212.6	216.6	220.4	224.0	227.4	230.7	233.9	237.0	240.0	242.9	245.7	248.2	31
32	22	207.5	211.7	215.7	219.5	223.1	226.5	220.8	233.0	236.1	239.1	242.0	244.8	247.3	32
33	23.03-23°B	203.3	207.5	211.5	215.3	218.9	222.3	225.6	228.8	231.9	234.9	237.8	240.6	243.1	33
34	24	200.1	204.2	208.3	212.1	215.7	219.1	222.4	225.6	228.7	231.7	234.6	237.4	240.0	34
35	24.99-24°B	196.5	200.7	204.7	208.5	212.1	215.5	218.8	222.0	225.1	228.1	231.0	233.8	236.3	35
36	26	193.3	197.5	201.6	205.3	208.9	212.2	215.6	218.9	221.9	225.0	237.8	230.6	233.1	36
37	27	190.2	194.3	198.4	202.2	205.7	209.0	212.5	215.8	218.7	221.8	234.7	227.4	230.0	37
38	27.66-25°B	187.6	191.8	195.8	199.6	203.2	206.6	209.9	213.1	216.2	219.2	222.1	224.9	227.4	38
39	28	186.6	190.7	194.8	198.5	202.2	205.6	208.8	212.1	215.1	218.2	221.0	223.9	226.3	39
40	29	183.5	187.6	191.8	195.4	199.1	202.6	205.7	209.0	212.1	215.1	217.9	220.9	223.2	40
41	29.60-26°B	181.4	185.6	189.6	193.4	197.0	200.4	203.7	206.9	210.0	213.0	215.9	218.7	221.2	41
42	30	180.2	184.4	188.4	192.2	195.8	199.2	202.5	205.7	208.8	211.8	214.7	217.5	220.0	42
43	31.05-27°B	177.0	181.2	185.2	189.0	192.6	196.0	199.3	202.5	206.6	209.6	212.5	215.3	217.8	43
44	32	174.4	178.6	182.6	186.4	190.0	193.4	196.7	199.9	204.0	207.0	209.9	212.7	215.2	44
45	33	171.7	175.9	179.9	183.7	187.3	190.7	194.0	197.2	201.3	204.3	207.2	210.0	212.5	45
46	33.25-28°B	170.8	175.0	179.0	182.8	186.4	189.8	193.1	196.3	200.4	203.4	206.3	209.1	211.6	46
47	34	168.9	173.1	177.1	180.9	184.5	187.9	191.2	195.4	198.5	201.5	204.4	207.2	209.7	47
48	35	166.3	170.5	174.5	178.3	181.9	185.3	188.6	192.8	195.9	198.9	201.8	204.6	207.1	48

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TABLE 18.—PROPERTIES OF AMMONIA LIQUOR (*Continued*)

(Starr's Table)

Showing the Relation Between Pressure and Temperature for Solutions of Ammonia in Water of Different Strengths

Line	Per Cent of NH_3 By Weight and Degree Beaumé	130	135	140	145	150	155	160	165	170	175	180	Line
1	1	348.8	351.3	353.7	356.0	358.2	360.3	362.3	364.2	366.1	367.4	369.5	1
2	1.84-11°B	344.4	346.9	349.3	351.6	353.8	355.9	357.9	359.8	361.7	363.5	365.1	2
3	2	343.6	346.1	348.2	350.8	352.9	355.1	357.1	358.9	360.9	362.7	364.3	3
4	3	338.3	340.8	342.9	345.5	347.6	349.8	351.8	353.6	355.6	357.4	359.0	4
5	3.80-12°B	333.7	336.2	338.6	340.9	343.1	345.2	347.2	349.2	351.1	352.8	354.4	5
6	4	332.6	335.2	337.6	339.8	342.1	344.1	346.2	348.2	350.1	351.7	353.4	6
7	5	327.3	329.9	332.3	334.5	336.8	338.8	340.9	342.9	344.9	346.4	348.1	7
8	5.30-13°B	325.9	328.4	330.8	333.1	335.3	337.4	339.4	341.3	343.2	345.0	346.6	8
9	6	322.4	324.8	327.3	329.5	331.8	333.8	335.9	337.7	339.6	341.4	343.7	9
10	6.80-14°B	317.8	320.3	322.7	325.0	327.2	329.3	331.3	333.3	335.2	336.9	338.5	10
11	7	316.9	319.3	321.8	324.0	321.3	328.3	330.4	332.3	334 1	335.9	336.6	11
12	8	312.0	314.3	316.9	319.1	321.4	327.3	325.5	327.4	329.2	330.9	331.7	12
13	8.22-15°B	309.8	312.3	314.7	317.0	319.2	321.3	323.3	325.3	327.2	328.9	330.5	13
14	9	306.4	308.9	311.3	313.6	315.8	317.9	319.9	321.8	323.7	325.5	327.2	14
15	10-16°B	301.1	303.6	306.0	308.3	310.5	312.6	314.6	316.6	318.5	320.2	321.8	15
16	11	296.6	299.2	301.5	303.8	305.0	308.2	310.2	312.1	314.0	315.8	317.5	16
17	12	292.1	294.7	297.0	299.3	300.5	303.8	305.8	307.6	309.5	311.3	313.0	17
18	12.17-17°B	290.1	292.6	295.0	297.3	299.5	301.6	303.6	305.6	307.5	309.2	310.8	18
19	13	286.5	289.1	291.5	293.7	296.0	298.1	300.1	302.0	304.0	305.7	307.3	19
20	13.88-18°B	282.9	285.4	287.8	290.1	292.3	294.4	296.4	298.3	300.1	302.0	303.6	20
21	14	282.4	284.9	287.3	289.6	291.8	293.9	295.9	297.8	299.6	301.5	303.1	21
22	15	278.2	280.7	283.1	285.4	287.6	289.7	291.7	293.6	295.4	297.3	298.9	22
23	16	274.0	276.5	279.9	281.2	283.4	285.5	287.5	289.4	291.3	293.1	294.7	23
24	16.22-19°B	272.8	275.3	277.7	280.0	282.2	284.3	286.3	288.3	290.3	292.0	293.6	24
25	17	269.6	272.1	274.5	276.8	278.0	281.1	283.1	285.1	287.1	288.8	290.4	25
26	18.03-20°B	265.2	267.7	270.1	272.4	274.6	276.7	278.7	280.7	282.6	284.3	285.9	26
27	19	261.2	263.8	266.1	268.5	270.6	272.8	274.7	276.8	278.6	280.4	282.0	27
28	19.87-21°B	258.0	260.5	262.9	265.2	267.4	269.5	271.5	273.5	275.4	277.1	278.7	28
29	20	257.5	260.1	262.4	264.7	266.9	269.0	271.1	273.0	275.0	276.6	278.2	29
30	21	253.8	256.4	259.7	261.0	263.2	265.3	267.4	269.3	271.3	272.9	274.5	30
31	21.75-22°B	250.8	253.3	255.7	258.0	260.2	262.3	264.3	266.3	268.2	269.9	271.5	31
32	22	249.9	252.4	254.8	257.1	259.3	261.4	263.4	265.4	267.3	269.0	270.6	32
33	23.03-23°B	245.7	248.2	250.6	252.9	255.1	257.2	259.2	261.2	263.1	264.8	266.4	33
34	24	242.6	246.0	247.4	249.7	251.9	254.0	256.0	257.9	259.8	261.5	263.1	34
35	24.99-24°B	238.9	241.4	243.8	246.1	248.3	250.4	252.4	254.4	256.3	258.0	259.6	35
36	26	235.8	238.3	240.6	242.9	245.2	247.3	249.3	251.2	253.1	254.9	256.5	36
37	27	232.6	235.2	237.4	239.8	242.1	244.2	246.1	248.0	249.9	251.7	253.4	37
38	27.66-25°B	230.0	232.5	234.9	237.2	239.4	241.5	243.5	245.3	247.2	248.9	250.5	38
39	28	229.0	231.4	233.9	236.1	238.4	240.4	242.5	244.3	246.1	247.8	246.5	39
40	29	225.9	228.4	230.8	233.0	235.4	237.3	239.4	241.3	243.1	244.8	245.5	40
41	29.60-26°B	223.8	226.3	228.7	231.0	233.2	235.3	237.3	239.3	241.2	242.9	244.5	41
42	30	222.6	225.1	227.5	229.8	232.0	234.1	236.1	238.1	240.0	241.7	243.3	42
43	31.05-27°B	220.4	222.9	225.3	227.6	229.8	231.9	233.9	235.8	237.5	239.2	240.8	43
44	32	217.8	220.3	222.7	225.0	227.2	229.3	231.3	233.2	234.9	236.5	238.2	44
45	33	215.1	217.6	220.0	222.7	224.5	226.6	228.6	230.5	232.2	233.8	235.5	45
46	33.25-28°B	214.2	216.7	219.1	221.8	223.6	225.6	227.5	229.3	231.1	232.8	234.4	46
47	34	212.3	214.8	217.2	219.9	221.7	223.7	225.6	227.4	229.2	230.9	232.5	47
48	35	209.7	212.2	214.6	217.3	219.1	221.1	223.8	224.8	226.8	228.2	231.3	48

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TABLE 19.—TEMPERATURE CORRECTIONS OF AQUA AMMONIA

The figures in top row indicate degrees Fahrenheit; those in the columns beneath give the strength of Ammonia at 60 degrees.

Degrees Beaumé	60°	66°	67°	68°	72°	74°	76°	78°	81°	84°	88°	90°	92°	95°	96°	100°	102°	108°	109°
18	18	....	....	17¾	....	....	17½	....	....	17¼	....	....	17	....	....	16¾	....	16½	....
18¼	18¼	....	....	18	....	....	17¾	....	....	17½	....	....	17¼	....	....	17	....	16¾	....
18½	18½	....	....	18¼	....	....	18	....	....	17¾	....	....	17½	....	....	17¼	....	17	....
18¾	18¾	....	....	18½	....	....	18¼	....	....	18	....	....	17¾	....	....	17½	....	17¼	....
19	19	....	....	18¾	....	....	18½	....	....	18¼	....	....	18	....	....	17¾	....	17½	....
19¼	19¼	....	....	19	....	....	18¾	....	....	18½	....	....	18¼	....	....	18	....	17¾	....
19½	19½	....	....	19¼	....	....	19	....	....	18¾	....	....	18½	....	....	18¼	....	18	....
19¾	19¾	....	....	19½	....	....	19¼	....	....	19	....	....	18¾	....	....	18½	....	18¼	....
20	20	....	19¾	....	....	19½	....	....	19¼	....	19	....	....	18¾	....	....	18½	....	18¼
20¼	20¼	....	20	....	....	19¾	....	....	19½	....	19¼	....	....	19	....	....	18¾	....	18½
20½	20½	....	20¼	....	....	20	....	....	19¾	....	19½	....	....	19¼	....	....	19	....	18¾
20¾	20¾	....	20½	....	....	20¼	....	....	20	....	19¾	....	....	19½	....	....	19¼	....	19
21	21	....	20¾	....	....	20½	....	....	20¼	....	20	....	....	19¾	....	....	19½	....	19¼
21¼	21¼	....	21	....	....	20¾	....	....	20½	....	20¼	....	....	20	....	....	19¾	....	19½
21½	21½	....	21¼	....	....	21	....	....	20¾	....	20½	....	....	20¼	....	....	20	....	19¾
21¾	21¾	....	21½	....	....	21¼	....	....	21	....	20¾	....	....	20½	....	....	20¼	....	20
22	22	21¾	....	....	21½	....	....	21¼	....	21	....	20¾	....	....	20½	....	20¼	20	....
22¼	22¼	22	....	....	21¾	....	....	21½	....	21¼	....	21	....	....	20¾	....	20½	20¼	....
22½	22½	22¼	....	....	22	....	....	21¾	....	21½	....	21¼	....	....	21	....	20¾	20½	....
22¾	22¾	22½	....	....	22¼	....	....	22	....	21¾	....	21½	....	....	21¼	....	21	20¾	....
23	23	22¾	....	....	22½	....	....	22¼	....	22	....	21¾	....	....	21½	....	21¼	21	....
23¼	23¼	23	....	....	22¾	....	....	22½	....	22¼	....	22	....	....	21¾	....	21½	21¼	....
23½	23½	23¼	....	....	23	....	....	22¾	....	22½	....	22¼	....	....	22	....	21¾	21½	....

TABLE 20.—AQUA AMMONIA

Be°	Sp. Gr.	%NH₃	Be°	Sp. Gr.	%NH₃	Be°	Sp. Gr.	%NH₃
10.00	1.0000	.00	16.50	.9556	11.18	23.00	.9150	23.52
10.25	.9982	.40	16.75	.9540	11.64	23.25	.9135	24.01
10.50	.9964	.80	17.00	.9524	12.10	23.50	.9121	24.50
10.75	.9947	1.21	17.25	.9508	12.56	23.75	.9106	24.99
11.00	.9929	1.62	17.50	.9492	13.02	24.00	.9091	25.48
11.25	.9912	2.04	17.75	.9475	13.49	24.25	.9076	25.97
11.50	.9894	2.46	18.00	.9459	13.96	24.50	.9061	26.46
11.75	.9876	2.88	18.25	.9444	14.43	24.75	.9047	26.95
12.00	.9859	3.30	18.50	.9428	14.90	25.00	.9032	27.44
12.25	.9842	3.73	18.75	.9412	15.37	25.25	.9018	27.93
12.50	.9825	4.16	19.00	.9396	15.84	25.50	.9003	28.42
12.75	.9807	4.59	19.25	.9380	16.32	25.75	.8989	28.91
13.00	.9790	5.02	19.50	.9365	16.80	26.00	.8974	29.40
13.25	.9773	5.45	19.75	.9349	17.28	26.25	.8960	29.89
13.50	.9756	5.88	20.00	.9333	17.76	26.50	.8946	30.38
13.75	.9739	6.31	20.25	.9318	18.24	26.75	.8931	30.87
14.00	.9722	6.74	20.50	.9302	18.72	27.00	.8917	31.36
14.25	.9705	7.17	20.75	.9287	19.20	27.25	.8903	31.85
14.50	.9689	7.61	21.00	.9272	19.68	27.50	.8889	32.34
14.75	.9672	8.05	21.25	.9256	20.16	27.75	.8875	32.83
15.00	.9655	8.49	21.50	.9241	20.64	28.00	.8861	33.32
15.25	.9639	8.93	21.75	.9226	21.12	28.25	.8847	33.81
15.50	.9622	9.38	22.00	.9211	21.60	28.50	.8833	34.30
15.75	.9605	9.83	22.25	.9195	22.08	28.75	.8819	34.79
16.00	.9589	10.28	22.50	.9180	22.56	29.00	.8805	35.28
16.25	.9573	10.73	22.75	.9165	23.04			

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TABLE 21.—PROPERTIES OF SOLUTIONS OF CHLORIDE OF SODIUM (COMMON SALT)

Percentage of Salt by Weight	Pounds of Salt Per Gallon of Solution	Degrees on Salometer at 60° F.	Weight Per Gallon at 39° F. =4°C.	Specific Gravity at 39° F. =4°C.	Specific Heat	Freezing Point F.
1	0.084	4	8.40	1.007	0.992	30.5
2	0.169	8	8.46	1.015	0.984	29.3
2.5	0.212	10	8.50	1.019	0.980	28.6
3	0.256	12	8.53	1.023	0.976	27.8
3.5	0.300	14	8.56	1.026	0.972	27.1
4	0.344	16	8.59	1.030	0.968	26.6
5	0.433	20	8.65	1.037	0.960	25.2
6	0.523	24	8.72	1.045	0.946	23.9
7	0.617	28	8.78	1.053	0.932	22.5
8	0.708	32	8.85	1.061	0.919	21.2
9	0.802	36	8.91	1.068	0.905	19.9
10	0.897	40	8.97	1.076	0.892	18.7
12	1.092	48	9.10	1.091	0.874	16.0
15	1.389	60	9.26	1.115	0.855	12.2
20	1.928	80	9.64	1.155	0.829	6.1
24	2.376	96	9.90	1.187	0.795	1.2
25	2.488	100	9.97	1.196	0.783	0 5
26	2.610	104	10.04	1.204	0.771	—1 1

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TABLE 22.—COMPARISONS OF THERMOMETER SCALES

+ Means above Zero, and − Means below Zero on each Scale. R. Reaumur or German Scale = 4/9 (F. − 32) = 4/5 C. C. Centigrade or French Scale = 3/9 (F. − 32) = 5/4 R. F. Fahrenheit or English Scale = 9/5 (C. + 32) = 9/4 R. + 32

R.	C.	F.	R	C.	F.
+80	+100	+212	+23	+28.75	+83.75
79	98 75	209.75	22	27 50	81.50
78	97 50	207 50	21	26.25	79 25
77	96.25	205.25	20	25	77
76	95	203	19	23.75	74.75
75	93.75	200.75	18	22.50	72.50
74	92.50	198.50	17	21.25	70 25
73	91.25	196 25	16	20	68
72	90	194	15	18.75	65.75
71	88.75	191.75	14	17 50	63.50
70	87.50	189 50	13	16.25	61.25
69	86.25	187 25	12	15	59
68	85	185	11	13.75	56.75
67	83 75	182.75	10	12.50	54.50
66	82.50	180 50	9	11.25	52.25
65	81.25	178 25	8	10	50
64	80	176	7	8.75	47.75
63	78.75	173.75	6	7.50	45.50
62	77 50	171.50	5	6.25	43.25
61	76.25	169 25	4	5	41
60	75	167	3	3.75	38.75
59	73 75	164 75	2	2.50	36.50
58	72 50	162 50	1	1.25	34.25
57	71.25	160 25	0	0	32
56	70	158	-1	-1.25	29 75
55	68 75	158 75	2	2.50	27.50
54	67 50	153 50	3	3.75	25.25
53	66 25	151 25	4	5	23
52	65	149	5	6.25	20.75
51	63 75	146 75	6	7.50	18.50
50	62.50	144 50	7	8.75	16.25
49	61.25	142 25	8	10	14
48	60	140	9	11.25	11.75
47	58 75	137.75	10	12 50	9.50
46	57 50	135 50	11	13 75	7.25
45	56.25	133 25	12	15	5
44	55	131	13	16.25	2.75
43	53 75	128 75	14	17.50	0.50
42	52.50	126 50	15	18.75	—1.75
41	51 25	124 25	16	20	4
40	50	122	17	21.25	6.25
39	48 75	119 75	18	22.50	8.50
38	47 50	117.50	19	23.75	10.75
37	46.25	115.25	20	25	13
36	45	113	21	26.25	15.25
35	43.75	110 75	22	27.50	17.50
34	42 50	109.50	23	28.75	19.75
33	41.25	106 25	24	30	22
32	40	104	25	31.25	24.25
31	38 75	101 75	26	32.50	26.50
30	37 50	99.50	27	33 75	28 75
29	36.25	97.25	28	35	31
28	35	95	29	36.25	33.25
27	33.75	92.75	30	37.50	35.50
26	32.50	90.50	31	38.75	37.75
25	31.25	88.25	32	40	40
24	30	86			

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TABLE 23.—SATURATED AMMONIA: PRESSURES

Pressure, lb. abs.	Temp., °F.	Volume, cu. ft. per lb.	Weight, lb. per cu. ft.	Heat content in B.t.u.		Latent heat in B.t.u.		Energy of vapor B.t.u.	Entropy.		
				of liquid	of vapor	of vaporisation	Internal		of liquid	of vaporisation	of vapor
p	t	v	1/v	h	H	L	ρ	u	n	L/T	N
1	−103.7	225.0	0.0044	.....	....	644.6	603.0	....	.....	1.8107	.....
2	− 87.1	117.0	.0085	.....	....	633.9	590.7	....	.....	1.7017	.....
3	− 76.5	80.0	.0125	.....	....	626.9	582.7	....	.....	1.6363	.....
4	− 68.5	61.0	.0164	.....	....	621.6	576.5	....	.....	1.5891	.....
5	− 62.0	49.3	0.0203	−98.1	519.1	617.2	571.5	473.4	−0.2207	1.5523	1.3316
6	− 56.6	41.6	.0241	−92.5	521.1	613.5	567.3	474.9	− .2070	1.5223	1.3153
7	− 51.9	35.9	.0279	−87.6	522.7	610.2	563.7	476.1	− .1947	1.4965	1.3018
8	− 47.6	31.6	.0316	−83.2	524.1	607.3	560.4	477.1	− .1840	1.4740	1.2900
9	− 43.9	28.3	.0352	−79.3	525.3	604.7	557.4	478.0	− .1747	1.4541	1.2794
10	− 40.4	25.75	0.0388	−75.7	526.4	602.2	554.6	478.8	−0.1661	1.4363	1.2702
11	− 37.2	23.60	.0424	−72.4	527.4	599.9	552.0	479.5	− .1584	1.4202	1.2618
12	− 34.3	21.75	.0460	−69.4	528.4	597.8	549.6	480.2	− .1513	1.4054	1.2541
13	− 31.5	20.16	.0496	−66.5	529.3	595.8	547.4	480.9	− .1446	1.3917	1.2471
14	− 28.9	18.79	.0532	−63.8	530.1	593.9	545.3	481.5	− .1384	1.3789	1.2405
15	− 26.4	17.60	0.0568	−61.2	530.9	592.1	543.3	482.0	−0.1324	1.3669	1.2345
16	− 24.1	16.56	.0604	−58.8	531.6	590.4	541.4	482.5	− .1268	1.3557	1.2289
17	− 21.9	15.64	.0639	−56.5	532.2	588.8	539.6	483.0	− .1215	1.3451	1.2236
18	− 19.8	14.82	.0675	−54.4	532.8	587.2	537.9	483.4	− .1165	1.3351	1.2186
19	− 17.8	14.09	.0710	−52.3	533.4	585.7	536.3	483.9	− .1119	1.3257	1.2138
20	− 15.9	13.45	0.0744	−50.3	534.0	584.3	534.7	484.3	−0.1075	1.3168	1.2093
21	− 14.0	12.82	.0780	−48.4	534.6	582.9	533.1	484.7	− .1032	1.3082	1.2050
22	− 12.2	12.27	.0815	−46.5	535.1	581.5	531.6	485.1	− .0990	1.2999	1.2009
23	− 10.5	11.77	.0850	−44.7	535.6	580.2	530.2	485.4	− .0950	1.2920	1.1970
24	− 8.8	11.30	.0885	−42.9	536.1	579.0	528.8	485.8	− .0912	1.2844	1.1932
25	− 7.2	10.88	0.0919	−41.3	536.5	577.8	527.4	486.1	−0.0876	1.2771	1.1896
26	− 5.7	10.50	.0953	−39.7	536.9	576.6	526.1	486.4	− .0840	1.2701	1.1862
27	− 4.2	10.13	.0987	−38.1	537.4	575.4	524.9	486.7	− .0805	1.2634	1.1829
28	− 2.7	9.78	.1022	−36.5	537.8	574.3	523.7	487.0	− .0771	1.2569	1.1798
29	− 1.3	9.47	.1056	−35.0	538.2	573.2	522.5	487.3	− .0739	1.2507	1.1768
30	+ 0.1	9.17	0.1090	−33.6	538.5	572.1	521.3	487.6	−0.0708	1.2447	1.1739
31	1.4	8.90	.1124	−32.2	538.9	571.1	520.2	487.8	− .0677	1.2389	1.1712
32	2.7	8.64	.1158	−30.8	539.3	570.1	519.1	488.1	− .0647	1.2332	1.1685
33	4.0	8.39	.1192	−29.5	539.6	569.1	518.0	488.4	− .0617	1.2275	1.1658
34	5.3	8.15	.1226	−28.2	540.0	568.1	516.9	488.6	− .0589	1.2220	1.1631
35	6.5	7.93	0.1260	−26.9	540.3	567.1	515.8	488.8	−0.0561	1.2167	1.1606
36	7.7	7.73	.1294	−25.6	540.6	566.2	514.8	489.1	− .0534	1.2116	1.1581
37	8.9	7.52	.1328	−24.4	540.9	565.3	513.8	489.3	− .0508	1.2066	1.1558
38	10.0	7.34	.1362	−23.2	541.2	564.4	512.8	489.5	− .0483	1.2018	1.1535
39	11.1	7.16	.1396	−22.0	541.5	563.5	511.9	489.7	− .0458	1.1971	1.1513
40	12.2	6.99	0.1430	−20.8	541.8	562.6	511.0	489.9	−0.0433	1.1924	1.1491
41	13.3	6.82	.1464	−19.7	542.0	561.7	510.0	490.1	− .0409	1.1878	1.1469
42	14.4	6.67	.1497	−18.6	542.3	560.9	509.1	490.3	− .0386	1.1833	1.1448
43	15.4	6.52	.1531	−17.5	542.6	560.0	508.2	490.5	− .0363	1.1790	1.1427
44	16.4	6.38	.1564	−16.4	542.8	559.2	507.3	490.7	− .0341	1.1748	1.1407
45	17.4	6.25	0.1598	−15.3	543.1	558.4	506.4	490.9	−0.0319	1.1707	1.1388
46	18.4	6.12	.1631	−14.3	543.3	557.6	505.6	491.1	− .0297	1.1666	1.1369
47	19.4	6.00	.1665	−13.3	543.6	556.8	504.7	491.3	− .0276	1.1626	1.1350
48	20.3	5.88	.1698	−12.3	543.8	556.1	503.9	491.4	− .0255	1.1587	1.1332
49	21.2	5.77	.1732	−11.3	544.0	555.3	503.1	491.6	− .0235	1.1549	1.1314
50	22.1	5.66	0.1765	−10.3	544.3	554.6	502.3	491.8	−0.0216	1.1512	1.1297
51	23.0	5.56	.1798	− 9.3	544.5	553.9	501.5	491.9	− .0196	1.1476	1.1280
52	23.9	5.46	.1831	− 8.4	544.7	553.1	500.8	492.1	− .0177	1.1441	1.1264
53	24.8	5.36	.1865	− 7.5	544.9	552.4	500.0	492.3	− .0158	1.1406	1.1247
54	25.6	5.27	.1898	− 6.6	545.1	551.7	499.3	492.4	− .0140	1.1372	1.1231

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TABLE 23.—SATURATED AMMONIA: PRESSURES (*Continued*)

Pressure, lb. abs.	Temp., °F.	Volume, cu. ft. per lb.	Weight, lb. per cu. ft.	Heat content in B.t.u.		Latent heat in B.t.u.		Energy of vapor B.t.u.	Entropy		
				of liquid	of vapor	of vaporization	Internal		of liquid	of vaporization	of vapor
p	t	v	1/v	h	H	L	ρ	u	n	L/T	N
55	26.4	5.28	0.1931	−5.7	545.3	551.1	498.6	492.6	−0.0122	1.1338	1.1216
56	27.3	5.09	.1964	−4.8	545.5	550.4	497.8	492.7	.0103	1.1304	1.1201
57	28.1	5.01	.1997	−3.9	545.7	549.7	497.1	492.9	.0085	1.1271	1.1186
58	28.9	4.93	.2030	−3.0	545.9	549.0	496.4	493.0	.0068	1.1239	1.1171
59	29.7	4.85	.2063	−2.2	546.1	548.4	495.7	493.2	.0050	1.1207	1.1157
60	30.5	4.77	0.2096	−1.3	546.3	547.7	495.0	493.3	−0.0033	1.1175	1.1142
61	31.3	4.70	.2129	−0.5	546.5	547.0	494.3	493.5	−0.0016	1.1144	1.1128
62	32.1	4.63	.2162	+0.3	546.7	546.4	493.6	493.6	+0.0001	1.1113	1.1114
63	32.8	4.56	.2195	1.1	546.9	545.8	492.9	493.7	.0018	1.1083	1.1101
64	33.6	4.49	.2228	1.9	547.1	545.1	492.2	493.9	.0035	1.1053	1.1087
65	34.3	4.42	0.2261	2.7	547.2	544.5	491.6	494.0	0.0051	1.1023	1.1074
66	35.1	4.36	.2294	3.5	547.4	543.8	490.9	494.1	.0067	1.0994	1.1061
67	35.8	4.30	.2327	4.3	547.6	543.2	490.3	494.3	.0082	1.0966	1.1048
68	36.5	4.24	.2359	5.1	547.7	542.6	489.6	494.4	.0097	1.0939	1.1036
69	37.2	4.18	.2392	5.8	547.9	542.0	489.0	494.5	.0113	1.0911	1.1024
70	37.9	4.12	0.2425	6.6	548.1	541.4	488.4	494.6	0.0128	1.0883	1.1011
71	38.6	4.07	.2458	7.4	548.2	540.8	487.7	494.8	.0143	1.0856	1.0999
72	39.3	4.02	.2490	8.1	548.4	540.2	487.1	494.9	.0158	1.0829	1.0987
73	40.0	3.96	.2523	8.9	548.5	539.6	486.5	495.0	.0173	1.0802	1.0975
74	40.7	3.91	.2556	9.6	548.7	539.0	485.9	495.1	.0187	1.0776	1.0963
75	41.3	3.86	0.2589	10.3	548.8	538.5	485.3	495.2	0.0201	1.0751	1.0952
76	42.0	3.81	.2622	11.0	549.0	537.9	484.7	495.3	.0215	1.0726	1.0941
77	42.6	3.77	.2655	11.7	549.1	537.4	484.1	495.4	.0229	1.0701	1.0930
78	43.3	3.72	.2688	12.4	549.3	536.8	483.5	495.6	.0243	1.0676	1.0919
79	43.9	3.68	.2721	13.1	549.4	536.3	482.9	495.7	.0257	1.0651	1.0908
80	44.5	3.63	0.2751	13.8	549.5	535.8	482.3	495.8	0.0271	1.0627	1.0898
81	45.1	3.59	.2786	14.5	549.7	535.2	481.8	495.9	.0284	1.0605	1.0888
82	45.8	3.55	.2819	15.2	549.8	534.6	481.2	496.0	.0297	1.0580	1.0877
83	46.4	3.51	.2851	15.8	550.0	534.1	480.6	496.1	.0310	1.0557	1.0867
84	47.0	3.47	.2884	16.5	550.1	533.6	480.1	496.2	.0323	1.0534	1.0857
85	47.6	3.43	0.2917	17.2	550.2	533.1	479.5	496.3	0.0336	1.0511	1.0847
86	48.2	3.39	.2950	17.8	550.4	532.5	479.0	496.4	.0349	1.0488	1.0837
87	48.8	3.35	.2983	18.5	550.5	532.0	478.4	496.5	.0362	1.0465	1.0827
88	49.4	3.32	.3015	19.1	550.6	531.5	477.9	496.6	.0374	1.0444	1.0817
89	50.0	3.28	.3048	19.8	550.8	531.0	477.3	496.7	.0386	1.0421	1.0807
90	50.5	3.25	0.1081	20.4	550.9	530.5	476.8	496.8	0.0398	1.0400	1.0798
91	51.1	3.21	.3114	21.0	551.0	530.0	476.3	496.9	.0410	1.0379	1.0789
92	51.7	3.18	.3147	21.7	551.1	529.5	475.8	497.0	.0422	1.0358	1.0780
93	52.2	3.14	.3180	22.3	551.2	529.0	475.3	497.1	.0434	1.0337	1.0771
94	52.8	3.11	.3213	22.9	551.4	528.5	474.8	497.2	.0446	1.0316	1.0762
95	53.3	3.08	0.3246	23.5	551.5	528.0	474.3	497.3	0.0458	1.0295	1.0753
96	53.9	3.05	.3278	24.1	551.6	527.5	473.8	497.4	.0470	1.0274	1.0744
97	54.4	3.02	.3311	24.7	551.7	527.0	473.3	497.4	.0482	1.0254	1.0736
98	55.0	2.99	.3344	25.3	551.9	526.5	472.8	497.5	.0494	1.0234	1.0727
99	55.5	2.96	.3377	25.9	552.0	526.1	472.3	497.6	.0505	1.0214	1.0719
100	56.0	2.93	0.3409	26.5	552.1	525.6	471.8	497.7	0.0516	1.0195	1.0710
101	56.6	2.90	.3442	27.1	552.2	525.1	471.3	497.8	.0527	1.0175	1.0702
102	57.1	2.88	.3475	27.7	552.3	524.6	470.8	497.9	.0539	1.0155	1.0694
103	57.6	2.85	.1508	28.2	552.4	524.2	470.3	498.0	.0550	1.0136	1.0686
104	58.1	2.82	.3540	28.8	552.5	523.7	469.8	498.1	.0561	1.0117	1.0678
105	58.6	2.80	0.3573	29.3	552.6	523.3	469.3	498.1	0.0572	1.0098	1.0670
106	59.1	2.77	.3605	29.9	552.7	522.8	468.9	498.2	.0583	1.0079	1.0662
107	59.6	2.75	.3638	30.4	552.8	522.4	468.4	498.3	.0594	1.0061	1.0654
108	60.1	2.72	.3670	31.0	552.9	521.9	467.9	498.4	.0605	1.0043	1.0647
109	60.6	2.70	.3703	31.5	553.0	521.5	467.5	498.5	.0614	1.0025	1.0639

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TABLE 23.—SATURATED AMMONIA: PRESSURES (*Continued*)

Pressure, lb. abs.	Temp., °F.	Volume, cu. ft. per lb.	Weight, lb. per cu. ft.	Heat content in B.t.u.		Latent heat in B.t.u.		Energy of vapor B.t.u.	Entropy		
				of liquid	of vapor	of vaporization	Internal		of liquid	of vaporization	of vapor
p	t	v	1/v	h	H	L	ρ	u	n	L/T	N
110	61.1	2.678	0.3735	32.1	553.1	521.0	467.0	498.6	0.0625	1.0006	1.0631
111	61.6	2.655	.3767	32.6	553.2	520.6	466.5	498.6	.0636	0.9988	1.0624
112	62.1	2.632	.3799	33.2	553.3	520.1	466.1	498.7	.0646	.9970	1.0616
113	62.6	2.610	.3831	33.7	553.4	519.7	465.6	498.8	.0657	.9951	1.0608
114	63.1	2.589	.3863	34.3	553.5	519.2	465.1	498.9	.0668	.9933	1.0601
115	63.6	2.568	0.3895	34.8	553.6	518.8	464.6	499.0	0.0678	0.9916	1.0594
116	64.0	2.547	.3927	35.4	553.7	518.4	464.2	499.0	.0688	.9899	1.0587
117	64.5	2.526	.3959	35.9	553.8	517.9	463.8	499.1	.0697	.9883	1.0580
118	64.9	2.506	.3991	36.4	553.9	517.5	463.4	499.2	.0706	.9867	1.0573
119	65.4	2.486	.4023	36.9	554.0	517.1	462.9	499.3	.0716	.9850	1.0566
120	65.8	2.466	0.4056	37.4	554.1	516.7	462.5	499.4	0.0725	0.9834	1.0559
121	66.3	2.446	.4089	37.9	554.2	516.3	462.1	499.4	.0735	.9817	1.0552
122	66.8	2.427	.4121	38.5	554.3	515.8	461.6	499.5	.0745	.9800	1.0545
123	67.2	2.409	.4153	39.0	554.4	515.4	461.2	499.6	.0754	.9784	1.0539
124	67.7	2.390	.4185	39.5	554.5	515.0	460.8	499.7	.0764	.9768	1.0532
125	68.1	2.371	0.4218	40.0	554.6	514.6	460.4	499.7	0.0773	0.9752	1.0525
126	68.6	2.353	.4250	40.5	554.7	514.1	459.9	499.8	.0783	.9736	1.0519
127	69.0	2.335	.4283	41.0	554.8	513.7	459.5	499.9	.0792	.9720	1.0512
128	69.5	2.317	.4316	41.5	554.9	513.3	459.0	500.0	.0802	.9704	1.0505
129	69.9	2.300	.4348	42.0	555.0	512.9	458.6	500.0	.0811	.9688	1.0499
130	70.4	2.283	0.4381	42.5	555.0	512.5	458.2	500.1	0.0820	0.9672	1.0492
131	70.8	2.266	.4414	43.0	555.1	512.1	457.8	500.2	.0829	.9656	1.0485
132	71.2	2.249	.4447	43.5	555.2	511.7	457.4	500.2	.0838	.9641	1.0479
133	71.6	2.233	.4479	44.0	555.3	511.3	457.0	500.3	.0847	.9626	1.0473
134	72.0	2.217	.4511	44.5	555.4	510.9	456.6	500.4	.0856	.9611	1.0467
135	72.5	2.201	0.4544	45.0	555.5	510.5	456.2	500.5	0.0865	0.9596	1.0461
136	72.9	2.185	.4577	45.5	555.6	510.1	455.8	500.5	.0874	.9581	1.0455
137	73.3	2.169	.4610	46.0	555.6	509.7	455.4	500.6	.0883	.9566	1.0449
138	73.7	2.154	.4643	46.4	555.7	509.4	455.0	500.7	.0892	.9551	1.0443
139	74.1	2.139	.4675	46.9	555.8	509.0	454.6	500.7	.0901	.9536	1.0437
140	74.5	2.124	0.4707	47.3	555.9	508.6	454.2	500.8	0.0910	0.9521	1.0431
141	75.0	2.109	.4740	47.8	556.0	508.2	453.8	500.9	.0919	.9506	1.0425
142	75.4	2.095	.4772	48.3	556.1	507.8	453.4	500.9	.0928	.9491	1.0419
143	75.8	2.082	.4804	48.8	556.1	507.4	453.0	501.0	.0936	.9477	1.0413
144	76.2	2.069	.4835	49.2	556.2	507.0	452.6	501.1	.0944	.9463	1.0407
145	76.5	2.056	0.4867	49.6	556.3	506.7	452.2	501.1	0.0952	0.9450	1.0402
146	76.9	2.043	.4899	50.0	556.4	506.3	451.8	501.2	.0960	.9436	1.0396
147	77.3	2.029	.4931	50.5	556.4	506.0	451.4	501.2	.0968	.9423	1.0391
148	77.7	2.015	.4963	50.9	556.5	505.6	451.0	501.3	.0976	.9410	1.0386
149	78.1	2.002	.4995	51.4	556.6	505.2	450.7	501.4	.0985	.9396	1.0380
150	78.5	1.989	0.5028	51.8	556.7	504.8	450.3	501.4	0.0993	0.9382	1.0375
151	78.9	1.976	.5060	52.3	556.7	504.4	449.9	501.5	.1002	.9368	1.0369
152	79.3	1.964	.5092	52.7	556.8	504.0	449.5	501.6	.1010	.9354	1.0364
153	79.6	1.952	.5123	53.1	556.9	503.7	449.2	501.6	.1018	.9341	1.0359
154	80.0	1.940	.5155	53.6	557.0	503.3	448.8	501.7	.1026	.9328	1.0354
155	80.4	1.928	0.5187	54.0	557.0	503.0	448.4	501.7	0.1034	0.9314	1.0348
156	80.8	1.916	.5220	54.5	557.1	502.6	448.0	501.8	.1042	.9301	1.0343
157	81.2	1.904	.5253	54.9	557.2	502.2	447.6	501.8	.1050	.9288	1.0338
158	81.5	1.892	.5286	55.3	557.2	501.9	447.3	501.9	.1058	.9275	1.0333
159	81.9	1.880	.5320	55.8	557.3	501.5	446.9	502.0	.1066	.9262	1.0328
160	82.3	1.868	0.5353	56.2	557.4	501.1	446.6	502.1	0.1074	0.9249	1.0323
161	82.7	1.857	.5386	56.7	557.5	500.7	446.2	502.1	.1082	.9236	1.0318
162	83.0	1.846	.5418	57.1	557.5	500.4	445.9	502.2	.1090	.9223	1.0313
163	83.4	1.835	.5450	57.5	557.6	500.0	445.5	502.2	.1098	.9210	1.0308
164	83.8	1.824	.5483	58.0	557.7	499.7	445.1	502.3	.1106	.9197	1.0303

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TABLE 23.—SATURATED AMMONIA: PRESSURES (*Continued*)

Pressure, lb. per sq. in. abs.	Temp., °F.	Volume, cu. ft. per lb.	Weight, lb. per cu. ft.	Heat content in B.t.u.		Latent heat in B.t.u.		Energy of vapor B.t.u.	Entropy		
				of liquid	of vapor	of vaporization	Internal		of liquid	of vaporization	of vapor
p	t	v	1/v	h	H	L	ρ	u	n	L/T	N
165	84.1	1.814	0.5515	58.4	557.7	499.4	444.8	502.3	0.1114	0.9184	1.0298
166	84.5	1.803	.5547	58.8	557.8	499.0	444.4	502.4	.1122	.9171	1.0293
167	84.9	1.793	.5578	59.3	557.9	498.6	444.0	502.5	.1130	.9158	1.0288
168	85.2	1.783	.5609	59.7	558.0	498.3	443.7	502.5	.1137	.9146	1.0283
169	85.6	1.773	.5641	60.1	558.0	497.9	443.3	502.6	.1145	.9133	1.0278
170	85.9	1.763	0.5673	60.5	558.1	497.6	443.0	502.7	0.1152	0.9121	1.0273
171	86.3	1.753	.5705	61.0	558.2	497.2	442.6	502.7	.1160	.9108	1.0268
172	86.6	1.743	.5738	61.4	558.2	496.9	442.3	502.8	.1167	.9096	1.0264
173	87.0	1.733	.5771	61.8	558.3	496.5	441.9	502.8	.1175	.9084	1.0259
174	87.3	1.723	.5804	62.2	558.4	496.2	441.6	502.9	.1182	.9072	1.0254
175	87.7	1.713	0.5836	62.6	558.4	495.8	441.2	502.9	0.1190	0.9060	1.0250
176	88.0	1.704	.5869	63.0	558.5	495.5	440.9	503.0	.1197	.9048	1.0245
177	88.4	1.694	.5902	63.4	558.6	495.1	440.5	503.0	.1204	.9037	1.0241
178	88.7	1.685	.5935	63.8	558.6	494.8	440.2	503.1	.1211	.9025	1.0236
179	89.0	1.676	.5967	64.2	558.7	494.5	439.9	503.1	.1218	.9013	1.0232
180	89.4	1.666	0.6000	64.6	558.8	494.1	439.5	503.2	0.1226	0.9001	1.0227
181	89.7	1.656	.6034	65.0	558.8	493.8	439.2	503.3	.1233	.8989	1.0223
182	90.1	1.647	.6068	65.4	558.9	493.4	438.8	503.3	.1241	.8977	1.0218
183	90.4	1.639	.6102	65.8	558.9	493.1	438.5	503.4	.1248	.8966	1.0214
184	90.7	1.630	.6135	66.2	559.0	492.8	438.2	503.4	.1254	.8955	1.0210
185	91.1	1.621	0.6168	66.6	559.1	492.4	437.8	503.5	0.1261	0.8944	1.0205
186	91.4	1.613	.6200	67.0	559.1	492.1	437.5	503.5	.1268	.8933	1.0201
187	91.7	1.605	.6233	67.4	559.2	491.8	437.2	503.6	.1274	.8921	1.0196
188	92.1	1.596	.6266	67.8	559.3	491.5	436.8	503.6	.1283	.8909	1.0192
189	92.4	1.588	.6298	68.2	559.3	491.2	436.5	503.7	.1289	.8898	1.0187
190	92.7	1.580	0.6330	68.6	559.4	490.9	436.2	503.7	0.1296	0.8887	1.0183
191	93.0	1.572	.6362	68.9	559.4	490.5	435.9	503.8	.1303	.8876	1.0179
192	93.4	1.563	.6395	69.3	559.5	490.1	435.5	503.9	.1310	.8865	1.0174
193	93.7	1.555	.6428	69.7	559.6	489.8	435.2	503.9	.1317	.8854	1.0170
194	94.0	1.548	.6460	70.1	559.6	489.5	434.9	504.0	.1323	.8843	1.0166
195	94.3	1.541	0.649	70.5	559.7	489.2	434.5	504.0	0.1329	0.8833	1.0162
196	94.6	1.533	.652	70.8	559.7	488.9	434.2	504.1	.1336	.8822	1.0158
197	94.9	1.526	.655	71.2	559.8	488.6	433.9	504.1	.1342	.8812	1.0154
198	95.2	1.519	.658	71.6	559.8	488.3	433.6	504.2	.1349	.8801	1.0150
199	95.5	1.512	.661	71.9	559.9	488.0	433.3	504.2	.1356	.8790	1.0146
200	95.9	1.504	0.665	72.3	560.0	487.6	433.0	504.3	0.1363	0.8779	1.0142
202	96.5	1.489	.672	73.1	560.1	487.0	432.3	504.4	.1376	.8758	1.0134
204	97.1	1.474	.679	73.8	560.2	486.4	431.7	504.5	.1389	.8737	1.0126
206	97.7	1.460	.685	74.6	560.3	485.8	431.1	504.6	.1402	.8716	1.0118
208	98.3	1.447	.691	75.3	560.4	485.1	430.5	504.7	.1414	.8696	1.0110
210	98.9	1.433	0.698	76.0	560.5	484.5	429.8	504.8	0.1427	0.8676	1.0103
212	99.5	1.419	.705	76.7	560.6	484.0	429.2	504.9	.1440	.8656	1.0095
214	100.1	1.406	.711	77.4	560.7	483.3	428.6	505.0	.1452	.8636	1.0088
216	100.7	1.394	.717	78.1	560.8	482.7	428.0	505.0	.1464	.8616	1.0080
218	101.2	1.382	.724	78.8	560.9	482.1	427.4	505.1	.1476	.8597	1.0073
220	101.8	1.370	0.730	79.5	561.0	481.5	426.8	505.2	0.1488	0.8578	1.0066
222	102.4	1.358	.736	80.2	561.1	480.9	426.2	505.3	.1500	.8559	1.0059
224	103.0	1.346	.743	80.9	561.2	480.3	425.6	505.4	.1512	.8540	1.0052
226	103.5	1.335	.749	81.6	561.3	479.7	425.1	505.5	.1524	.8521	1.0045
228	104.1	1.323	.756	82.3	561.4	479.1	424.5	505.6	.1537	.8501	1.0038
230	104.7	1.312	0.762	83.0	561.5	478.5	423.9	505.7	0.1549	0.8482	1.0031
232	105.2	1.301	.769	83.7	561.6	477.9	423.3	505.8	.1561	.8463	1.0024
234	105.8	1.290	.775	84.4	561.7	477.3	422.7	505.9	.1573	.8444	1.0017
236	106.3	1.279	.782	85.0	561.8	476.8	422.2	505.9	.1585	.8426	1.0011
238	106.9	1.268	.789	85.7	561.9	476.2	421.6	506.0	.1597	.8407	1.0004

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TABLE 23.—SATURATED AMMONIA: PRESSURES (*Continued*)

Pressure, lb. abs.	Temp., °F.	Volume, cu. ft. per lb.	Weight, lb. per cu. ft.	Heat content in B.t.u.		Latent heat in B.t.u.		Energy of vapor B.t.u.	Entropy		
				of liquid	of vapor	of vaporization	Internal		of liquid	of vaporization	of vapor
p	t	v	1/v	h	H	L	ρ	u	n	1/T	N
240	107.4	1.258	0.795	86.4	562.0	475.6	421.0	506.1	0.1609	0.8389	0.9998
242	108.0	1.248	.801	87.1	562.1	475.0	420.4	506.2	.1621	.8371	.9991
244	108.5	1.238	.808	87.7	562.2	474.5	419.8	506.3	.1632	.8353	.9985
246	109.0	1.228	.814	88.4	562.3	473.9	419.3	506.4	.1643	.8335	.9979
248	109.6	1.218	.821	89.1	562.4	473.3	418.7	506.5	.1655	.8317	.9972
250	110.1	1.208	0.828	89.7	562.5	472.8	418.1	506.6	0.1666	0.8300	0.9966
252	110.6	1.199	.834	90.4	562.6	472.2	417.6	506.6	.1677	.8283	.9960
254	111.1	1.189	.841	91.0	562.6	471.6	417.1	506.7	.1688	.8266	.9954
256	111.7	1.179	.848	91.7	562.7	471.0	416.5	506.8	.1700	.8248	.9947
258	112.2	1.170	.855	92.3	562.8	470.5	415.9	506.9	.1711	.8231	.9941
260	112.7	1.161	0.861	93.0	562.9	470.0	415.4	507.0	0.1722	0.8213	0.9935
262	113.2	1.153	.867	93.6	563.0	469.4	414.8	507.1	.1733	.8196	.9929
264	113.7	1.144	.874	94.2	563.1	468.9	414.3	507.2	.1744	.8179	.9923
266	114.2	1.136	.880	94.8	563.2	468.3	413.8	507.2	.1755	.8162	.9917
268	114.7	1.127	.887	95.5	563.3	467.8	413.2	507.3	.1766	.8145	.9911
270	115.2	1.119	0.894	96.1	563.4	467.2	412.7	507.4	0.1777	0.8129	0.9906
272	115.7	1.110	.901	96.7	563.4	466.7	412.2	507.5	.1787	.8113	.9900
274	116.2	1.102	.908	97.4	563.5	466.1	411.7	507.6	.1798	.8096	.9894
276	116.7	1.094	.914	98.0	563.6	465.6	411.2	507.7	.1809	.8080	.9888
278	117.1	1.087	.920	98.6	563.7	465.1	410.7	507.7	.1819	.8064	.9883
280	117.6	1.079	0.927	99.2	563.8	464.6	410.2	507.8	0.1829	0.8048	0.9878
282	118.1	1.071	.934	99.8	563.9	464.0	409.6	507.9	.1840	.8032	.9872
284	118.6	1.063	.941	100.4	563.9	463.5	409.1	508.0	.1850	.8016	.9867
286	119.1	1.056	.947	101.1	564.0	462.9	408.5	508.1	.1861	.8000	.9861
288	119.6	1.049	.953	101.7	564.1	462.4	408.0	508.2	.1872	.7984	.9856
290	120.0	1.042	0.960	102.3	564.2	461.9	407.5	508.2	0.1882	0.7969	0.9851
292	120.5	1.035	.966	102.9	564.3	461.4	407.0	508.3	.1892	.7954	.9846
294	120.9	1.028	.973	103.5	564.3	460.9	406.5	508.4	.1902	.7939	.9841
296	121.4	1.021	.980	104.1	564.4	460.4	406.0	508.5	.1912	.7924	.9836
298	121.9	1.014	.986	104.7	564.5	459.8	405.5	508.6	.1922	.7908	.9830
300	122.4	1.007	0.993	105.3	564.6	459.3	405.0	508.7	0.1932	0.7893	0.9825
310	124.6	0.975	1.026	108.2	565.0	456.8	402.5	509.0	.1981	.7820	.9801
320	126.8	.945	1.059	111.1	565.3	454.3	400.0	509.4	.2030	.7747	.9777
330	129.0	.916	1.092	114.0	565.7	451.8	397.6	509.8	.2078	.7676	.9754
340	131.1	.889	1.125	116.8	566.1	449.3	395.2	510.1	.2125	.7606	.9731
350	133.2	0.863	1.159	119.6	566.4	446.8	392.8	510.5	0.2171	0.7538	0.9709
360	135.2	.838	1.193	122.3	566.7	444.4	390.5	510.8	.2216	.7472	.9688
370	137.2	.815	1.227	125.0	567.0	442.0	388.2	511.2	.2261	.7407	.9668
380	139.2	.793	1.261	127.7	567.3	439.6	385.9	511.5	.2305	.7343	.9648
390	141.1	.772	1.295	130.3	567.6	437.3	383.7	511.9	.2348	.7281	.9629
400	142.9	0.752	1.330	132.9	567.9	435.0	381.5	512.2	0.2390	0.7220	0.9610
410	144.8	.733	1.364	135.5	568.2	432.7	379.3	512.5	.2431	.7161	.9592
420	146.6	.715	1.399	138.1	568.5	430.4	377.2	512.8	.2472	.7102	.9574
430	148.4	.698	1.434	140.6	568.8	428.2	375.0	513.2	.2513	.7044	.9557
440	150.1	.681	1.469	143.1	569.0	426.0	372.9	513.5	.2553	.6987	.9540
450	151.8	0.665	1.504	145.6	569.3	423.8	370.8	513.8	0.2593	0.6931	0.9524
460	153.5	.650	1.539	148.0	569.6	421.6	368.7	514.1	.2632	.6876	.9508
470	155.2	.636	1.574	150.4	569.8	419.4	366.6	514.4	.2671	.6822	.9493
480	156.9	.622	1.608	152.8	570.1	417.2	364.5	514.7	.2710	.6768	.9478
490	158.5	.609	1.642	155.2	570.3	415.0	362.5	515.0	.2748	.6715	.9464
500	160.0	0.597	1.675	157.5	570.5	413.0	360.5	515.3	0.2786	0.6664	0.9450
525	163.9	.566	1.765	163.4	571.1	407.7	355.6	516.0	.2876	.6540	.9416
550	167.6	.539	1.855	169.2	571.7	402.5	350.8	516.7	.2965	.6419	.9384
575	171.2	.514	1.946	174.8	572.2	397.4	346.0	517.4	.3052	.6301	.9353
600	174.7	0.491	2.038	180.4	572.7	392.3	341.3	518.1	0.3138	0.6186	0.9324
625	178.1	.469	2.132	185.9	573.1	387.2	336.6	518.8	.3223	.6073	.9296
650	181.4	.449	2.227	191.4	573.6	382.2	332.0	519.5	.3307	.5963	.9270
675	184.6	.431	2.321	196.8	574.0	377.2	327.4	520.1	.3389	.5856	.9245
700	187.7	.414	2.416	202.1	574.4	372.2	322.8	520.7	.3469	.5752	.9221

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TABLE 24.—SATURATED AMMONIA: TEMPERATURES

Temp. °F. abs.	Pressure, lb. per sq. in.	Volume, cu. ft. per lb.	Weight, lb. per cu. ft.	Heat content in B.t.u.		Latent heat in B.t.u.		Energy of vapor B.t.u.	Entropy		
				of liquid	of vapor	of vaporization	Internal		of liquid	of vaporization	of vapor
t	p	v	1/v	h	H	L	ρ	u	η	L/T	N
−50	7.43	34.01	0.02940	−85.7	523.3	608.9	562.2	476.5	−0.1901	1.4866	1.2965
−49	7.67	32.98	.03032	−84.7	523.6	608.2	561.4	476.7	− .1875	1.4812	1.2937
−48	7.92	32.00	.03125	−83.6	523.9	607.5	560.6	476.9	− .1850	1.4759	1.2909
−47	8.17	31.07	.03218	−82.6	524.3	606.8	559.8	477.2	− .1825	1.4706	1.2881
−46	8.43	30.19	.03312	−81.5	524.6	606.1	559.0	477.4	− .1801	1.4654	1.2854
−45	8.69	29.34	0.03408	−80.5	524.9	605.4	558.2	477.6	−0.1776	1.4602	1.2826
−44	8.96	28.52	.03506	−79.5	525.3	604.7	557.4	477.9	− .1751	1.4550	1.2798
−43	9.24	27.72	.03607	−78.4	525.6	604.0	556.6	478.2	− .1727	1.4498	1.2771
−42	9.53	26.94	.03712	−77.4	525.9	603.3	555.8	478.4	− .1702	1.4446	1.2744
−41	9.82	26.18	.03820	−76.4	526.2	602.6	555.0	478.6	− .1678	1.4395	1.2717
−40	10.12	25.45	0.03930	−75.3	526.6	601.9	554.2	478.9	−0.1653	1.4344	1.2691
−39	10.43	24.74	.04042	−74.3	526.9	601.2	553.4	479.1	− .1628	1.4293	1.2664
−38	10.75	24.06	.04156	−73.3	527.2	600.5	552.6	479.3	− .1604	1.4242	1.2638
−37	11.07	23.40	.04273	−72.2	527.5	599.7	551.8	479.6	− .1580	1.4191	1.2612
−36	11.40	22.76	.04393	−71.2	527.9	599.0	551.0	479.8	− .1555	1.4141	1.2586
−35	11.74	22.14	0.04516	−70.2	528.2	598.3	550.2	480.0	−0.1531	1.4091	1.2560
−34	12.09	21.55	.04641	−69.1	528.5	597.6	549.4	480.3	− .1507	1.4041	1.2534
−33	12.45	20.97	.04769	−68.1	528.8	596.9	548.6	480.5	− .1483	1.3991	1.2508
−32	12.81	20.41	.04900	−67.1	529.1	596.1	547.8	480.7	− .1458	1.3941	1.2483
−31	13.18	19.87	05033	−66.0	529.4	595.4	547.0	481.0	− .1434	1.3892	1.2458
−30	13.56	19.35	0.05168	−65.0	529.8	594.7	546.2	481.2	−0.1410	1.3843	1.2433
−29	13.95	18.84	05306	−63.9	530.1	594.0	545.4	481.4	− .1386	1.3794	1.2408
−28	14.35	18.35	05449	−62.9	530.4	593.2	544.6	481.6	− .1362	1.3745	1.2383
−27	14.76	17.87	05596	−61.8	530.7	592.5	543.7	481.9	− .1338	1.3697	1.2359
−26	15.18	17.40	05747	−60.8	531.0	591.8	542.9	482.1	− .1314	1.3648	1.2334
−25	15.61	16.95	0.0590	−59.8	531.3	591.1	542.1	482.3	−0.1290	1.3599	1.2310
−24	16.05	16.51	0606	−58.7	531.6	590.3	541.3	482.5	− .1266	1.3551	1.2286
−23	16.50	16.09	0622	−57.7	531.9	589.6	540.5	482.8	− .1242	1.3503	1.2261
−22	16.96	15.68	0638	−56.6	532.2	588.8	539.7	483.0	− .1218	1.3455	1.2237
−21	17.43	15.28	0654	−55.6	532.5	588.1	538.9	483.2	− .1195	1.3408	1.2214
−20	17.91	14.89	0.0671	−54.6	532.8	587.4	538.0	483.4	−0.1171	1.3361	1.2190
−19	18.40	14.52	.0689	−53.5	533.1	586.6	537.2	483.6	− .1147	1.3314	1.2166
−18	18.90	14.16	.0706	−52.5	533.4	585.9	536.4	483.8	− .1124	1.3267	1.2143
−17	19.41	13.81	.0724	−51.4	533.7	585.1	535.6	484.0	− .1100	1.3220	1.2119
−16	19.93	13.48	.0742	−50.4	534.0	584.4	534.8	484.3	− .1077	1.3173	1.2096
−15	20.46	13.15	0.0760	−49.4	534.3	583.6	513.9	484.5	−0.1054	1.3127	1.2073
−14	21.00	12.83	.0779	−48.3	534.6	582.9	533.1	484.7	− .1031	1.3081	1.2050
−13	21.56	12.51	.0799	−47.3	534.8	582.1	532.3	484.9	− .1007	1.3034	1.2027
−12	22.13	12.21	.0819	−46.2	535.1	581.4	531.4	485.1	− .0984	1.2988	1.2004
−11	22.71	11.92	.0839	−45.2	535.4	580.6	530.6	485.3	− .0961	1.2942	1.1981
−10	23.30	11.63	0.0860	−44.2	535.7	579.9	529.8	485.5	−0.0938	1.2897	1.1959
− 9	23.90	11.35	.0881	−43.1	536.0	579.1	528.9	485.7	− .0915	1.2852	1.1936
− 8	24.52	11.08	.0903	−42.1	536.3	578.4	528.1	485.9	− .0893	1.2807	1.1914
− 7	25.15	10.82	.0924	−41.0	536.6	577.6	527.3	486.1	− .0870	1.2762	1.1892
− 6	25.80	10.57	.0946	−40.0	536.9	576.8	526.4	486.3	− .0847	1.2717	1.1870
− 5	26.46	10.32	0.0969	−38.9	537.1	576.1	525.6	486.6	−0.0824	1.2672	1.1848
− 4	27.13	10.08	.0992	−37.9	537.4	575.3	524.7	486.8	− .0801	1.2627	1.1826
− 3	27.82	9.85	1015	−36.8	537.7	574.6	523.9	487.0	− .0778	1.2583	1.1805
− 2	28.52	9.62	1039	−35.8	538.0	573.8	523.1	487.2	− .0755	1.2538	1.1783
− 1	29.23	9.40	1064	−34.7	538.2	573.0	522.2	487.4	− .0732	1.2494	1.1762
0	29.95	9.19	0.1089	−33.7	538.5	572.2	521.4	487.6	−0.0709	1.2450	1.1741
1	30.69	8.98	.1114	−32.6	538.8	571.4	520.5	487.8	− .0686	1.2406	1.1720
2	31.44	8.78	.1139	−31.6	539.1	570.7	519.7	488.0	− .0663	1.2363	1.1700
3	32.21	8.58	1165	−30.5	539.3	569.9	518.8	488.2	− .0640	1.2319	1.1679
4	32.99	8.39	1192	−29.5	539.6	569.1	518.0	488.4	− .0618	1.2276	1.1658

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TABLE 24.—SATURATED AMMONIA: TEMPERATURES (*Continued*)

Temp., °F. abs.	Pressure, lb. per sq. in.	Volume, cu. ft. per lb.	Weight, lb. per cu. ft.	Heat content in B.t.u.		Latent heat in B.t.u.		Energy of vapor B.t.u.	Entropy		
				of liquid	of vapor	of vaporization	Internal		of liquid	of vaporization	of vapor
t	p	v	1/v	h	H	L	ρ	u	n	L/T	N
5	33.79	8.20	0.1219	−28.4	539.9	568.3	517.1	488.6	−0.0595	1.2232	1.1637
6	34.60	8.02	.1247	−27.4	540.1	567.5	516.3	488.7	− .0572	1.2189	1.1616
7	35.43	7.84	.1275	−26.3	540.4	566.7	515.4	488.9	− .0550	1.2146	1.1596
8	36.28	7.67	.1304	−25.3	540.7	565.9	514.6	489.1	− .0527	1.2103	1.1576
9	37.14	7.50	.1333	−24.2	540.9	565.2	513.7	489.3	− .0505	1.2060	1.1555
10	38.02	7.34	0.1363	−23.2	541.2	564.4	512.9	489.5	−0.0483	1.2018	1.1535
11	38.92	7.18	.1393	−22.1	541.4	563.6	512.0	489.7	− .0461	1.1976	1.1515
12	39.84	7.02	.1424	−21.1	541.7	562.8	511.2	489.9	− .0438	1.1933	1.1495
13	40.77	6.87	.1455	−20.0	542.0	562.0	510.3	490.1	− .0416	1.1891	1.1475
14	41.71	6.72	.1487	−19.0	542.2	561.2	509.4	490.3	− .0394	1.1849	1.1455
15	42.67	6.583	0.1519	−17.9	542.5	560.4	508.6	490.5	−0.0372	1.1807	1.1435
16	43.65	6.444	.1552	−16.8	542.7	559.6	507.7	490.6	− .0350	1.1766	1.1416
17	44.65	6.308	.1585	−15.8	543.0	558.8	506.8	490.8	− .0328	1.1724	1.1396
18	45.67	6.176	.1619	−14.7	543.2	558.0	506.0	491.0	− .0306	1.1683	1.1377
19	46.70	6.047	.1654	−13.6	543.5	557.1	505.1	491.2	− .0284	1.1641	1.1357
20	47.75	5.920	0.1689	−12.6	543.7	556.3	504.2	491.4	−0.0262	1.1600	1.1338
21	48.82	5.796	.1725	−11.5	544.0	555.5	503.3	491.6	− .0240	1.1559	1.1319
22	49.91	5.676	.1762	−10.4	544.2	554.7	502.4	491.8	− .0218	1.1517	1.1299
23	51.02	5.560	.1799	− 9.4	544.5	553.9	501.6	491.9	− .0196	1.1476	1.1280
24	52.15	5.447	.1836	− 8.3	544.7	553.1	500.7	492.1	− .0174	1.1436	1.1261
25	53.30	5.336	0.1874	− 7.3	545.0	552.2	499.8	492.3	−0.0153	1.1396	1.1243
26	54.47	5.228	.1913	− 6.2	545.2	551.4	498.9	492.5	− .0131	1.1355	1.1224
27	55.66	5.122	.1953	− 5.1	545.5	550.6	498.1	492.7	− .0109	1.1315	1.1206
28	56.87	5.019	.1993	− 4.1	545.7	549.8	497.2	492.9	− .0087	1.1275	1.1188
29	58.10	4.918	.2034	− 3.0	546.0	549.0	496.3	493.0	− .0066	1.1235	1.1169
30	59.35	4.820	0.2075	− 1.9	546.2	548.1	495.4	493.2	−0.0044	1.1195	1.1151
31	60.62	4.724	.2117	− 0.8	546.4	547.3	494.5	493.4	− .0022	1.1155	1.1133
32	61.91	4.631	.2159	+ 0.3	546.7	546.5	493.6	493.6	.0000	1.1115	1.1115
33	63.22	4.540	.2203	+ 1.3	546.9	545.6	492.8	493.8	+ .0021	1.1076	1.1098
34	64.55	4.451	.2247	+ 2.4	547.1	544.8	491.9	493.9	+ .0043	1.1037	1.1080
35	65.91	4.364	0.2292	3.5	547.4	543.9	491.0	494.1	0.0065	1.0997	1.1062
36	67.29	4.279	.2337	4.6	547.6	543.1	490.1	494.3	.0087	1.0958	1.1044
37	68.69	4.196	.2384	5.6	547.8	542.2	489.2	494.5	.0108	1.0919	1.1027
38	70.11	4.115	.2431	6.7	548.1	541.4	488.3	494.6	.0130	1.0880	1.1010
39	71.56	4.036	.2478	7.8	548.3	540.5	487.4	494.8	.0151	1.0841	1.0992
40	73.03	3.959	0.2526	8.9	548.5	539.7	486.5	495.0	0.0173	1.0802	1.0975
41	74.53	3.884	.2575	10.0	548.8	538.8	485.5	495.2	.0194	1.0764	1.0958
42	76.05	3.810	.2625	11.1	549.0	537.9	484.6	495.3	.0216	1.0725	1.0941
43	77.59	3.738	.2675	12.2	549.2	537.1	483.7	495.5	.0237	1.0687	1.0924
44	79.16	3.668	.2727	13.3	549.4	536.2	482.8	495.7	.0259	1.0648	1.0907
45	80.75	3.599	0.2779	14.3	549.7	535.3	481.9	495.9	0.0280	1.0610	1.0890
46	82.37	3.532	.2832	15.4	549.9	534.5	481.0	496.0	.0301	1.0572	1.0873
47	84.01	3.466	.2885	16.5	550.1	533.6	480.1	496.2	.0323	1.0534	1.0857
48	85.68	3.402	.2940	17.6	550.3	532.7	479.2	496.4	.0344	1.0496	1.0840
49	87.37	3.339	.2995	18.7	550.6	531.8	478.3	496.5	.0366	1.0458	1.0823
50	89.09	3.278	0.3051	19.8	550.8	531.0	477.3	496.7	0.0387	1.0420	1.0807
51	90.83	3.219	.3107	20.9	551.0	530.1	476.4	496.9	.0408	1.0383	1.0791
52	92.59	3.161	.3164	22.0	551.2	529.2	475.5	497.0	.0430	1.0345	1.0775
53	94.38	3.104	.3222	23.1	551.4	528.3	474.6	497.2	.0451	1.0307	1.0758
54	96.19	3.048	.3281	24.2	551.6	527.4	473.6	497.4	.0473	1.0269	1.0742
55	98.0	2.992	0.3342	25.3	551.9	526.5	472.7	497.5	0.0494	1.0232	1.0726
56	99.9	2.938	.3404	26.4	552.1	525.6	471.8	497.7	.0516	1.0195	1.0711
57	101.8	2.885	.3467	27.5	552.3	524.7	470.8	497.9	.0537	1.0158	1.0695
58	103.7	2.833	.3530	28.7	552.5	523.8	469.9	498.0	.0559	1.0121	1.0679
59	105.7	2.783	.3594	29.8	552.7	522.9	469.0	498.2	.0580	1.0084	1.0664

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TABLE 24.—SATURATED AMMONIA: TEMPERATURES (*Continued*)

Temp., °F. abs.	Pressure, lb. per sq. in.	Volume, cu. ft. per lb.	Weight, lb. per cu. ft.	Heat content in B.t.u.		Latent heat in B.t.u.		Energy of vapor B.t.u.	Entropy		
				of liquid	of vapor	of vaporization	Internal		of liquid	of vaporisation	of vapor
t.	p	v	1/v	h	H	L	ρ	u	η	L/T	N
60	107.7	2.734	0.3658	30.9	552.9	522.0	468.0	498.4	0.0601	1.0047	1.0648
61	109.7	2.686	.3723	32.0	553.1	521.1	467.1	498.5	.0623	1.0010	1.0633
62	111.7	2.639	.3790	33.1	553.3	520.2	466.1	498.7	.0644	0.9974	1.0618
63	113.8	2.592	.3858	34.2	553.5	519.3	465.2	498.9	.0665	.9938	1.0603
64	115.9	2.547	.3927	35.3	553.7	518.4	464.2	499.0	.0687	.9901	1.0588
65	118.1	2.503	0.3996	36.5	554.0	517.5	463.3	499.2	0.0708	0.9864	1.0572
66	120.3	2.460	.4066	37.6	554.2	516.5	462.3	499.4	.0729	.9828	1.0557
67	122.5	2.418	.4136	38.7	554.4	515.6	461.4	499.5	.0750	.9792	1.0542
68	124.7	2.377	.4207	39.9	554.6	514.7	460.4	499.7	.0771	.9756	1.0527
69	126.9	2.336	.4280	41.0	554.8	513.7	459.5	499.9	.0792	.9720	1.0512
70	129.2	2.296	0.4354	42.1	555.0	512.8	458.5	500.0	0.0813	0.9684	1.0497
71	131.5	2.257	.4430	43.3	555.2	511.9	457.6	500.2	.0834	.9648	1.0482
72	133.9	2.219	.4506	44.4	555.4	511.0	456.6	500.4	.0855	.9613	1.0468
73	136.3	2.182	.4583	45.5	555.6	510.0	455.7	500.5	.0876	.9577	1.0453
74	138.7	2.145	.4662	46.7	555.8	509.1	454.7	500.7	.0898	.9541	1.0439
75	141.1	2.109	0.4742	47.8	556.0	508.1	453.7	500.9	0.0919	0.9505	1.0424
76	143.6	2.074	.4823	49.0	556.2	507.2	452.7	501.0	.0940	.9470	1.0410
77	146.1	2.039	.4905	50.1	556.4	506.2	451.7	501.2	.0961	.9435	1.0396
78	148.7	2.005	.4988	51.3	556.6	505.3	450.8	501.3	.0983	.9399	1.0382
79	151.3	1.972	.5071	52.4	556.8	504.3	449.8	501.5	.1004	.9364	1.0368
80	153.9	1.940	0.5155	53.6	557.0	503.4	448.8	501.7	0.1025	0.9329	1.0354
81	156.5	1.908	.5241	54.8	557.1	502.4	447.8	501.8	.1047	.9294	1.0340
82	159.2	1.877	.5328	55.9	557.3	501.4	446.9	502.0	.1068	.9259	1.0327
83	161.9	1.847	.5416	57.1	557.5	500.5	445.9	502.2	.1090	.9223	1.0313
84	164.6	1.817	.5504	58.3	557.7	499.5	444.9	502.3	.1111	.9188	1.0299
85	167.4	1.788	0.5594	59.4	557.9	498.5	443.9	502.5	0.1132	0.9154	1.0286
86	170.2	1.759	.5685	60.6	558.1	497.5	442.9	502.7	.1153	.9119	1.0272
87	173.0	1.731	.5777	61.8	558.3	496.5	441.9	502.8	.1175	.9084	1.0259
88	175.9	1.704	.5870	63.0	558.5	495.5	440.9	503.0	.1196	.9050	1.0246
89	178.8	1.677	.5964	64.2	558.7	494.5	439.9	503.1	.1217	.9015	1.0232
90	181.8	1.650	0.6060	65.3	558.9	493.5	438.9	503.3	0.1238	0.8981	1.0219
91	184.8	1.624	.6158	66.5	559.1	492.5	437.9	503.5	.1259	.8946	1.0206
92	187.8	1.598	.6258	67.7	559.2	491.5	436.9	503.6	.1281	.8911	1.0192
93	190 9	1.573	.6358	68.9	559.4	490.5	435.9	503.8	.1302	.8877	1.0179
94	194.1	1.548	.6460	70.1	559.6	489.5	434.9	504.0	.1323	.8843	1.0166
95	197.3	1.524	0.656	71.3	559.8	488.5	433.9	504.1	0.1344	0.8809	1.0153
96	200.5	1.500	.667	72.5	560.0	487.5	432.8	504.3	.1365	.8775	1.0140
97	203.8	1.477	.677	73.7	560.2	486.5	431.8	504.5	.1387	.8741	1.0128
98	207.1	1.454	.688	74.9	560.3	485.4	430.8	504.6	.1408	.8707	1.0115
99	210.4	1.431	.699	76.1	560.5	484.4	429.8	504.8	.1429	.8673	1.0102
100	213.8	1.408	0.710	77.3	560.7	483.4	428.7	504.9	0.1450	0.8639	1.0089
101	217.2	1.386	.721	78.5	560.9	482.3	427.7	505.1	.1471	.8605	1.0077
102	220.7	1.365	.732	79.7	561.1	481.3	426.7	505.2	.1493	.8571	1.0064
103	224.2	1.345	.743	80.9	561.2	480.3	425.6	505.4	.1514	.8538	1.0052
104	227.7	1.325	.755	82.2	561.4	479.2	424.6	505.6	.1535	.8504	1.0040
105	231.2	1.305	0.766	83.4	561.6	478.2	423.5	505.7	0.1557	0.8470	1.0027
106	234.8	1.285	.778	84.6	561.8	477.1	422.5	505.9	.1578	.8437	1.0015
107	238.4	1.266	.790	85.8	561.9	476.1	421.4	506.1	.1599	.8404	1.0003
108	242.1	1.247	.802	87.1	562.1	475.0	420.4	506.2	.1621	.8370	0.9991
109	245.8	1.228	.814	88.3	562.3	474.0	419.3	506.4	.1642	.8337	0.9979
110	249.6	1.210	0.826	89.6	562.5	472.9	418.3	506.6	0.1664	0.8303	0.9967
111	253.4	1.192	.839	90.8	562.6	471.8	417.2	506.7	.1686	.8269	.9955
112	257.3	1.174	.852	92.1	562.8	470.7	416.1	506.9	.1707	.8236	.9943
113	261.2	1.156	.865	93.3	563.0	469.6	415.1	507.1	.1729	.8203	.9931
114	265.2	1.139	.878	94.6	563.1	468.5	414.0	507.2	.1750	.8170	.9920

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TABLE 24.—SATURATED AMMONIA: TEMPERATURES (*Continued*)

Temp., ° F. abs.	Pressure, lb. per sq. in.	Volume, cu. ft. per lb.	Weight, lb. per cu. ft.	Heat content in B.t.u.		Latent heat in B.t.u.		Energy of vapor B.t.u.	Entropy		
				of liquid	of vapor	of vaporization	Internal		of liquid	of vaporization	of vapor
t	p	v	1/v	h	H	L	ρ	u	η	L/T	N
115	269.2	1.122	0.891	95.9	563.3	467.4	412.9	507.4	0.1772	0.8136	0.9908
116	273.3	1.105	.905	97.1	563.5	466.3	411.9	507.6	.1794	.8103	.9897
117	277.4	1.088	.919	98.4	563.7	465.2	410.8	507.7	.1816	.8069	.9885
118	281.5	1.072	.933	99.7	563.8	464.1	409.7	507.9	.1838	.8036	.9874
119	285.7	1.057	.946	101.0	564.0	463.0	408.6	508.1	.1859	.8003	.9862
120	289.9	1.042	0.960	102.2	564.2	461.9	407.5	508.2	0.1881	0.7970	0.9851
121	294.2	1.027	0.974	103.5	564.4	460.8	406.4	508.4	.1903	.7937	.9840
122	298.5	1.012	0.988	104.8	564.5	459.7	405.3	508.6	.1925	.7904	.9829
123	302.8	0.998	1.002	106.1	564.7	458.6	404.2	508.8	.1947	.7871	.9818
124	307.2	0.984	1.016	107.4	564.9	457.4	403.1	508.9	.1968	.7839	.9807
125	311.6	0.970	1.031	108.7	565.0	456.3	402.0	509.1	0.1990	0.7806	0.9796
126	316.1	.956	1.046	110.0	565.2	455.2	400.9	509.3	.2012	.7773	.9785
127	320.6	.942	1.061	111.3	565.4	454.0	399.8	509.4	.2034	.7740	.9774
128	325.2	.929	1.076	112.6	565.5	452.8	398.7	509.6	.2056	.7708	.9764
129	329.9	.916	1.092	114.0	565.7	451.7	397.6	509.8	.2078	.7675	.9753
130	334.6	0.903	1.108	115.3	565.9	450.6	396.4	510.0	0.2100	0.7642	0.9742
131	339.4	.890	1.124	116.6	566.0	449.4	395.3	510.2	.2122	.7610	.9732
132	344.2	.877	1.140	118.0	566.2	448.2	394.2	510.3	.2145	.7577	.9721
133	349.0	.865	1.156	119.3	566.4	447.0	393.0	510.5	.2167	.7544	.9711
134	353.9	.853	1.172	120.7	566.5	445.8	391.9	510.7	.2189	.7512	.9701
135	358.8	0.841	1.189	122.0	566.7	444.7	390.7	510.8	0.2211	0.7479	0.9690
136	363.8	.829	1.206	123.4	566.8	443.5	389.6	511.0	.2234	.7446	.9680
137	368.9	.817	1.224	124.7	567.0	442.3	388.4	511.2	.2256	.7414	.9670
138	374.0	.806	1.241	126.1	567.2	441.1	387.3	511.3	.2278	.7381	.9659
139	379.2	.795	1.258	127.4	567.3	439.9	386.1	511.5	.2301	.7348	.9649
140	384.4	0.784	1.275	128.8	567.5	438.6	384.9	511.7	0.2323	0.7316	0.9639
141	389.7	.773	1.293	130.2	567.6	437.4	383.8	511.8	.2346	.7283	.9629
142	395.0	.762	1.312	131.6	567.8	436.2	382.6	512.0	.2368	.7251	.9619
143	400.4	.751	1.331	133.0	567.9	435.0	381.4	512.2	.2391	.7218	.9609
144	405.8	.741	1.349	134.4	568.1	433.7	380.2	512.4	.2413	.7186	.9599
145	411.3	0.731	1.368	135.8	568.3	432.5	379.0	512.6	0.2436	0.7154	0.9589
146	416.8	.721	1.387	137.2	568.4	431.2	377.8	512.7	.2459	.7121	.9579
147	422.4	.711	1.406	138.6	568.6	430.0	376.6	512.9	.2481	.7089	.9570
148	428.0	.701	1.425	140.0	568.7	428.7	375.4	513.1	.2504	.7056	.9560
149	433.7	.692	1.445	141.5	568.9	427.4	374.2	513.3	.2527	.7024	.9551
150	439.5	0.683	1.465	142.9	569.0	426.2	373.0	513.4	0.2550	0.6991	0.9541
155	469.1	.638	1.567	150.1	569.8	419.7	366.9	514.4	.2666	.6829	.9495
160	500.1	.597	1.676	157.5	570.5	413.0	360.6	515.3	.2784	.6666	.9450
165	532.6	.558	1.792	165.1	571.3	406.2	354.2	516.2	.2903	.6503	.9406
170	566.6	.522	1.915	172.9	572.0	399.1	347.6	517.2	.3023	.6340	.9363
175	602.2	0.489	2.045	180.9	572.7	391.8	340.8	518.2	0.3146	0.6175	0.9321
180	639.5	.458	2.183	189.1	573.4	384.3	333.9	519.2	.3271	.6010	.9281
185	678.4	.429	2.330	197.5	574.0	376.6	326.8	520.2	.3399	.5843	.9242
190	719.0	.402	2.488	206.2	574.7	368.5	319.4	521.2	.3530	.5674	.9204
195	761.4	.376	2.660	215.2	575.4	360.2	311.8	522.3	.3664	.5503	.9167

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TABLE 25.—SUPERHEATED AMMONIA

Pressure	15 [−26.4]			16 [−24.1]			17 [−21.9]			18 [−19.8]		
Temp °F	v	h	H_g	v	h	H_g	v	h	H_g	v	h	H_g
Sat.	17.6	1.234	530.9	16.6	1.229	531.6	15.6	1.224	532.2	14.8	1.219	532.8
−20	17.9	1.243	534.4	16.8	1.234	533.8	15.7	1.226	533.3	....	.....	.....
−10	18.4	1.255	539.9	17.3	1.246	539.3	16.2	1.238	538.9	15.3	1.231	538.4
0	18.9	1.267	545.2	17.7	1.258	544.7	16.6	1.250	544.3	15.7	1.243	543.9
10	19.3	1.278	550.5	18.1	1.270	550.1	17.0	1.262	549.7	16.1	1.255	549.3
20	19.8	1.289	555.7	18.6	1.281	555.4	17.4	1.273	555.0	16.4	1.266	554.7
30	20.2	1.300	560.9	19.0	1.292	560.6	17.8	1.284	560.3	16.8	1.277	560.0
40	20.7	1.310	566.0	19.4	1.302	565.8	18.2	1.295	565.5	17.2	1.288	565.2
50	21.1	1.320	571.1	19.8	1.312	570.9	18.6	1.305	570.6	17.6	1.298	570.3
60	21.5	1.330	576.2	20.2	1.322	576.0	19.0	1.315	575.7	18.0	1.308	575.5
70	22.0	1.340	581.3	20.6	1.332	581.0	19.4	1.325	580.8	18.3	1.318	580.6
80	22.4	1.349	586.3	21.0	1.343	586.0	19.8	1.334	585.8	18.7	1.327	585.7
90	22.9	1.358	591.4	21.5	1.350	591.1	20.2	1.343	591.0	19.1	1.336	590.8
100	23.3	1.367	596.4	21.9	1.359	596.1	20.6	1.352	596.0	19.4	1.345	595.8
110	23.7	1.376	601.4	22.3	1.368	601.1	21.0	1.361	601.0	19.8	1.354	600.8
120	24.2	1.385	606.4	22.7	1.377	606.2	21.4	1.370	606.0	20.1	1.363	605.9
130	24.6	1.394	611.4	23.1	1.386	611.2	21.7	1.379	611.0	20.5	1.372	610.9
140	25.1	1.402	616.4	23.5	1.394	616.2	22.1	1.387	616.1	20.9	1.380	616.0
150	25.5	1.410	621.4	23.9	1.403	621.2	22.5	1.395	621.1	21.2	1.388	621.0
160	25.9	1.418	626.4	24.3	1.411	626.2	22.9	1.404	626.1	21.6	1.397	626.0
170	26.3	1.426	631.4	24.7	1.419	631.2	23.3	1.412	631.1	22.0	1.405	631.0
180	26.8	1.434	636.4	25.1	1.427	636.3	23.7	1.420	636.2	22.4	1.413	636.1
200	27.7	1.449	646.5	26.0	1.442	646.4	24.5	1.435	646.3	23.2	1.428	646.2
220	28.5	1.464	656.7	26.8	1.457	656.6	25.3	1.450	656.5	23.9	1.443	656.4
240	29.3	1.479	666.9	27.6	1.472	666.8	26.1	1.465	666.7	24.7	1.458	666.6

	19 [−17.8]			20 [−15.9]			21 [−14.0]			22 [−12.2]		
Sat.	14.1	1.214	533.4	13.4	1.209	534.0	12.8	1.205	534.6	12.3	1.201	535.1
−10	14.4	1.224	537.9	13.7	1.217	537.4	13.0	1.210	536.9	12.4	1.204	536.4
0	14.8	1.236	543.4	14.0	1.229	542.9	13.3	1.222	542.5	12.7	1.216	542.1
10	15.2	1.248	548.8	14.4	1.241	548.4	13.7	1.234	548.1	13.1	1.228	547.7
20	15.5	1.259	554.2	14.7	1.252	553.9	14.0	1.246	553.6	13.4	1.240	553.2
30	15.9	1.270	559.6	15.1	1.263	559.2	14.4	1.257	558.9	13.7	1.251	558.6
40	16.3	1.281	564.9	15.5	1.274	564.5	14.7	1.268	564.2	14.0	1.262	564.0
50	16.7	1.291	570.1	15.8	1.284	569.8	15.0	1.278	569.5	14.3	1.272	569.3
60	17.0	1.301	575.3	16.2	1.294	575.0	15.4	1.288	574.7	14.7	1.282	574.5
70	17.3	1.311	580.4	16.5	1.304	580.2	15.7	1.298	579.9	15.0	1.292	579.7
80	17.7	1.320	585.5	16.8	1.314	585.3	16.0	1.308	585.1	15.3	1.302	584.9
90	18.0	1.329	590.6	17.1	1.323	590.4	16.3	1.317	590.2	15.6	1.312	590.0
100	18.4	1.339	595.7	17.5	1.332	595.5	16.6	1.326	595.3	15.9	1.321	595.1
110	18.7	1.348	600.7	17.8	1.341	600.5	16.9	1.335	600.4	16.2	1.330	600.2
120	19.0	1.357	605.8	18.1	1.350	605.6	17.2	1.344	605.4	16.5	1.339	605.3
130	19.4	1.365	610.8	18.4	1.359	610.7	17.5	1.353	610.5	16.8	1.347	610.4
140	19.8	1.374	615.8	18.8	1.368	615.7	17.8	1.362	615.5	17.1	1.356	615.4
150	20.1	1.382	620.9	19.1	1.376	620.8	18.1	1.370	620.6	17.3	1.364	620.5
160	20.4	1.390	625.9	19.4	1.384	625.8	18.4	1.378	625.7	17.6	1.372	625.6
170	20.8	1.398	630.9	19.7	1.392	630.8	18.7	1.386	630.7	17.9	1.381	630.6
180	21.2	1.406	636.0	20.1	1.400	635.9	19.1	1.394	635.8	18.2	1.389	635.7
200	21.9	1.421	646.1	20.8	1.415	646.0	19.7	1.410	645.9	18.8	1.404	645.8
220	22.6	1.437	656.3	21.4	1.431	656.2	20.3	1.425	656.1	19.4	1.419	656.0
240	23.2	1.452	666.5	22.0	1.446	666.4	20.9	1.440	666.4	20.0	1.434	666.2
260	23.8	1.467	676.7	22.6	1.461	676.6	21.5	1.455	676.5	20.6	1.449	676.5

v = volume, cu. ft. per lb. s = entropy i = heat content, B.t.u.

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TABLE 25.—SUPERHEATED AMMONIA (*Continued*)

Pressure	23 [−10.5]			24 [−8.8]			25 [−7.2]			26 [−5.7]		
Temp °F	v	N	Hs	v	N	Hs	v	N	Hs	v	N	Hs
Sat.	11.8	1.197	535.6	11.3	1.193	536.1	10.9	1.190	536.5	10.5	1.186	536.9
0	12.1	1.210	541.7	11.6	1.204	541.2	11.1	1.199	540.8	10.7	1.193	540.3
10	12.5	1.222	547.3	11.9	1.217	546.9	11.4	1.211	546.5	11.0	1.206	546.1
20	12.8	1.234	552.8	12.2	1.229	552.4	11.7	1.223	552.1	11.3	1.218	551.7
30	13.1	1.245	558.3	12.5	1.240	557.9	12.0	1.235	557.6	11.5	1.230	557.2
40	13.4	1.256	563.7	12.8	1.251	563.3	12.3	1.246	563.0	11.8	1.241	562.7
50	13.7	1.267	569.0	13.1	1.262	568.7	12.6	1.256	568.4	12.1	1.251	568.1
60	14.0	1.277	574.3	13.4	1.272	574.0	12.8	1.266	573.7	12.3	1.261	573.4
70	14.3	1.287	579.5	13.7	1.282	579.2	13.1	1.276	578.9	12.6	1.271	578.7
80	14.6	1.297	584.7	14.0	1.291	584.4	13.4	1.286	584.1	12.9	1.281	583.9
90	14.9	1.306	589.8	14.3	1.301	589.5	13.7	1.296	589.3	13.2	1.291	589.2
100	15.2	1.315	595.0	14.5	1.310	594.7	13.9	1.305	594.5	13.4	1.300	594.4
110	15.5	1.324	600.1	14.8	1.319	599.9	14.2	1.314	599.7	13.6	1.309	599.6
120	15.8	1.333	605.2	15.1	1.328	605.0	14.5	1.323	604.8	13.9	1.318	604.7
130	16.0	1.342	610.3	15.3	1.337	610.1	14.7	1.332	609.9	14.1	1.327	609.8
140	16.3	1.351	615.3	15.6	1.346	615.1	15.0	1.341	615.0	14.4	1.336	614.9
150	16.5	1.359	620.4	15.8	1.354	620.2	15.2	1.349	620.1	14.6	1.344	620.0
160	16.8	1.367	625.4	16.1	1.362	625.3	15.4	1.357	625.2	14.8	1.352	625.1
170	17.1	1.375	630.5	16.4	1.370	630.4	15.7	1.365	630.3	15.1	1.360	630.2
180	17.4	1.384	635.6	16.7	1.378	635.5	16.0	1.373	635.4	15.4	1.368	635.3
190	17.7	1.391	640.7	16.9	1.386	640.6	16.2	1.381	640.5	15.6	1.376	640.4
200	18.0	1.399	645.8	17.2	1.394	645.7	16.5	1.389	645.6	15.9	1.384	645.5
220	18.5	1.414	656.0	17.7	1.409	655.9	17.0	1.404	655.8	16.4	1.399	655.8
240	19.1	1.429	666.2	18.3	1.424	666.1	17.6	1.419	666.0	16.9	1.414	666.0
260	19.7	1.444	676.4	18.9	1.438	676.3	18.1	1.433	676.2	17.4	1.429	676.2

	27 [−4.2]			28 [−2.7]			30 [+0.1]			32 [2.7]		
Sat.	10.1	1.183	537.4	9.8	1.180	537.8	9.17	1.174	538.5	8.64	1.168	539.3
0	10.2	1.188	539.8	9.9	1.183	539.4	. . .	. . .	. . .	. . .	. . .	. . . . .
10	10.5	1.201	545.7	10.2	1.196	545.3	9.45	1.187	544.5	8.83	1.178	543.7
20	10.8	1.213	551.4	10.4	1.208	551.0	9.70	1.199	550.3	9.06	1.190	549.5
30	11.1	1.225	556.9	10.7	1.220	556.6	9.94	1.211	556.0	9.29	1.202	555.3
40	11.3	1.236	562.4	10.9	1.231	562.1	10.18	1.222	561.6	9.52	1.213	561.0
50	11.6	1.246	567.9	11.2	1.242	567.6	10.41	1.233	567.1	9.74	1.224	566.5
60	11.8	1.257	573.2	11.4	1.252	572.9	10.64	1.243	572.5	9.95	1.235	572.0
70	12.1	1.267	578.5	11.7	1.262	578.2	10.87	1.253	577.9	10.17	1.245	577.5
80	12.4	1.277	583.7	11.9	1.272	583.5	11.10	1.263	583.2	10.38	1.255	582.8
90	12.6	1.286	589.0	12.2	1.282	588.8	11.32	1.273	588.5	10.60	1.265	588.1
100	12.9	1.296	594.2	12.4	1.291	594.0	11.55	1.283	593.7	10.81	1.275	593.3
110	13.1	1.305	599.4	12.6	1.300	599.2	11.77	1.292	598.9	11.02	1.284	598.6
120	13.4	1.314	604.6	12.9	1.309	604.4	11.99	1.301	604.1	11.23	1.293	603.8
130	13.6	1.323	609.7	13.1	1.318	609.5	12.21	1.310	609.3	11.44	1.302	609.0
140	13.8	1.331	614.8	13.2	1.326	614.6	12.43	1.318	614.4	11.64	1.310	614.1
150	14.1	1.340	619.9	13.6	1.335	619.7	12.65	1.327	619.5	11.85	1.319	619.3
160	14.3	1.348	625.0	13.8	1.344	624.9	12.87	1.335	624.7	12.05	1.327	624.5
170	14.5	1.356	630.1	14.0	1.352	630.0	13.08	1.344	629.8	12.25	1.336	629.6
180	14.8	1.364	635.2	14.3	1.360	635.1	13.30	1.352	634.9	12.46	1.344	634.7
190	15.0	1.372	640.3	14.5	1.368	640.2	13.52	1.360	640.0	12.67	1.352	639.8
200	15.3	1.380	645.4	14.7	1.376	645.3	13.74	1.367	645.2	12.87	1.359	645.0
220	15.8	1.395	655.7	15.2	1.391	655.6	14.17	1.383	655.4	13.27	1.375	655.3
240	16.2	1.410	665.9	15.6	1.406	665.8	14.59	1.398	665.7	13.67	1.390	665.6
260	16.7	1.425	676.1	16.1	1.420	676.1	15.02	1.412	676.0	14.08	1.404	675.8

v = volume, cu. ft. per lb. s = entropy i = heat content, B.t.u.

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TABLE 25.—SUPERHEATED AMMONIA (*Continued*)

Pressure	34 [5.3]			36 [7.7]			38 [10.0]			40 [12.2]		
Temp °F.	v	N	H_S	v	N	H_S	v	N	H_S	v	N	H_S
Sat.	8.15	1.163	540.0	7.73	1.158	540.6	7.34	1.153	541.2	6.99	1.149	541.8
10	8.26	1.169	542.9	7.78	1.161	542.0	7.34	1.153	541.2	....	.....	.....
20	8.50	1.182	548.8	8.01	1.174	548.0	7.56	1.166	547.3	7.17	1.159	546.6
30	8.72	1.194	554.6	8.22	1.186	553.9	7.76	1.179	553.3	7.36	1.172	552.6
40	8.93	1.205	560.4	8.42	1.198	559.7	7.96	1.191	559.1	7.54	1.184	558.5
50	9.14	1.216	566.0	8.62	1.209	565.4	8.15	1.202	564.8	7.72	1.195	564.3
60	9.35	1.227	571.5	8.81	1.220	571.0	8.33	1.213	570.5	7.90	1.206	570.0
70	9.56	1.237	577.0	9.01	1.230	576.5	8.52	1.223	576.0	8.08	1.217	575.5
80	9.76	1.247	582.4	9.20	1.240	581.9	8.70	1.233	581.4	8.25	1.227	581.0
90	9.96	1.257	587.7	9.39	1.250	587.3	8.88	1.243	586.8	8.42	1.237	586.4
100	10.16	1.267	593.0	9.58	1.260	592.6	9.06	1.253	592.2	8.59	1.247	591.8
110	10.36	1.276	598.2	9.77	1.269	597.8	9.24	1.262	597.5	8.76	1.256	597.2
120	10.56	1.285	603.4	9.96	1.278	603.1	9.42	1.272	602.8	8.93	1.265	602.5
130	10.76	1.294	608.7	10.15	1.287	608.4	9.60	1.281	608.1	9.10	1.274	607.8
140	10.95	1.303	613.9	10.33	1.296	613.6	9.77	1.289	613.3	9.27	1.283	613.0
150	11.14	1.312	619.1	10.51	1.305	618.8	9.95	1.298	618.5	9.44	1.292	618.2
160	11.33	1.320	624.2	10.69	1.313	624.0	10.12	1.307	623.7	9.61	1.300	623.4
170	11.52	1.328	629.3	10.87	1.321	629.1	10.29	1.315	628.9	9.77	1.309	628.6
180	11.72	1.336	634.5	11.06	1.329	634.3	10.47	1.323	634.1	9.93	1.317	633.9
190	11.91	1.344	639.7	11.24	1.337	639.5	10.64	1.331	639.3	10.10	1.325	639.1
200	12.10	1.352	644.9	11.42	1.346	644.7	10.81	1.339	644.5	10.26	1.333	644.3
220	12.48	1.368	655.1	11.78	1.361	655.0	11.15	1.354	654.8	10.58	1.348	654.6
240	12.86	1.383	665.4	12.13	1.376	665.3	11.49	1.369	665.2	10.90	1.363	665.0
260	13.24	1.397	675.7	12.48	1.391	675.6	11.83	1.384	675.5	11.23	1.378	675.4
280	13.62	1.412	686.0	12.84	1.405	685.9	12.17	1.398	685.8	11.55	1.392	685.7

	42 [14.4]			44 [16.4]			46 [18.4]			48 [20.3]		
Sat.	6.67	1.145	542.3	6.38	1.141	542.8	6.12	1.137	543.3	5.88	1.133	543.8
20	6.81	1.152	545.8	6.47	1.145	545.1	6.15	1.139	544.3	....	.....	.....
30	7.00	1.165	551.9	6.65	1.158	551.3	6.33	1.152	550.6	6.06	1.146	550.0
40	7.17	1.177	557.9	6.82	1.170	557.3	6.51	1.164	556.7	6.22	1.158	556.1
50	7.34	1.188	563.7	6.99	1.182	563.2	6.68	1.176	562.6	6.38	1.170	562.0
60	7.51	1.199	569.5	7.16	1.193	569.0	6.84	1.187	568.4	6.54	1.182	567.9
70	7.68	1.210	575.1	7.32	1.204	574.6	6.99	1.198	574.1	6.69	1.193	573.6
80	7.85	1.220	580.6	7.48	1.214	580.2	7.14	1.209	579.7	6.84	1.203	579.3
90	8.02	1.230	586.0	7.64	1.224	585.7	7.29	1.219	585.3	6.98	1.213	584.9
100	8.18	1.240	591.5	7.80	1.234	591.1	7.44	1.229	590.8	7.13	1.223	590.4
110	8.34	1.250	596.9	7.95	1.244	596.5	7.59	1.238	596.2	7.27	1.233	595.9
120	8.50	1.259	602.2	8.10	1.253	601.9	7.74	1.248	601.6	7.42	1.242	601.3
130	8.66	1.268	607.5	8.26	1.262	607.2	7.89	1.257	606.9	7.56	1.251	606.6
140	8.82	1.277	612.7	8.41	1.271	612.5	8.04	1.266	612.2	7.70	1.260	611.9
150	8.98	1.286	618.0	8.56	1.280	617.7	8.18	1.275	617.5	7.84	1.269	617.2
160	9.14	1.294	623.2	8.71	1.289	623.0	8.32	1.283	622.8	7.98	1.278	622.5
170	9.30	1.303	628.4	8.86	1.297	628.2	8.47	1.292	628.0	8.12	1.286	627.8
180	9.45	1.311	633.7	9.01	1.305	633.5	8.61	1.300	633.3	8.25	1.295	633.1
190	9.61	1.319	638.9	9.16	1.313	638.7	8.76	1.308	638.5	8.39	1.303	638.3
200	9.76	1.327	644.1	9.31	1.321	643.9	8.90	1.316	643.7	8.53	1.311	643.5
210	9.92	1.335	649.3	9.46	1.329	649.1	9.04	1.324	648.9	8.67	1.319	648.8
220	10.08	1.342	654.5	9.61	1.337	654.3	9.18	1.331	654.1	8.80	1.326	654.0
240	10.38	1.357	664.9	9.90	1.352	664.7	9.46	1.346	664.6	9.06	1.341	664.5
260	10.69	1.372	675.3	10.19	1.366	675.1	9.74	1.361	675.0	9.33	1.356	674.9
280	10.99	1.386	685.7	10.48	1.381	685.6	10.02	1.375	685.5	9.60	1.370	685.4

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TABLE 25.—SUPERHEATED AMMONIA (*Continued*)

Pressure	50 [22.1]			55 [26.4]			60 [30.5]			65 [34.3]		
Temp °F.	v	N	H_S	v	N	H_S	v	N	H_S	v	N	H_S
Sat.	5.67	1.130	544.3	5.18	1.122	545.3	4.77	1.114	546.3	4.42	1.107	547.2
30	5.80	1.140	549.4	5.23	1.126	547.6	....	.....	.....	....	.....	.....
40	5.96	1.153	555.5	5.38	1.139	554.0	4.90	1.127	552.5	4.50	1.115	551.0
50	6.11	1.165	561.5	5.52	1.151	560.1	5.03	1.139	558.8	4.62	1.128	557.4
60	6.26	1.176	567.4	5.66	1.163	566.1	5.16	1.151	564.9	4.74	1.140	563.6
70	6.41	1.187	573.2	5.80	1.174	572.0	5.29	1.162	570.9	4.86	1.151	569.7
80	6.55	1.198	578.9	5.93	1.185	577.8	5.41	1.173	576.8	4.97	1.162	575.7
90	6.69	1.208	584.5	6.06	1.196	583.5	5.53	1.184	582.6	5.08	1.173	581.6
100	6.83	1.218	590.0	6.19	1.206	589.1	5.65	1.194	588.2	5.19	1.184	587.3
110	6.97	1.228	595.5	6.32	1.216	594.7	5.77	1.204	593.8	5.30	1.194	593.0
120	7.11	1.237	601.0	6.45	1.225	600.2	5.88	1.214	599.4	5.41	1.204	598.7
130	7.24	1.246	606.4	6.57	1.234	605.6	6.00	1.223	604.9	5.52	1.213	604.2
140	7.38	1.255	611.7	6.69	1.244	611.0	6.11	1.233	610.4	5.63	1.222	609.7
150	7.51	1.264	617.0	6.81	1.253	616.4	6.22	1.242	615.8	5.73	1.231	615.2
160	7.65	1.273	622.3	6.94	1.261	621.7	6.34	1.250	621.1	5.84	1.240	620.6
170	7.78	1.281	627.6	7.06	1.270	627.0	6.46	1.259	626.5	5.95	1.249	626.0
180	7.91	1.290	632.9	7.18	1.278	632.3	6.57	1.267	631.8	6.05	1.257	631.3
190	8.04	1.298	638.1	7.30	1.286	637.6	6.68	1.275	637.1	6.15	1.265	636.6
200	8.17	1.306	643.3	7.42	1.294	642.9	6.79	1.283	642.4	6.26	1.274	641.9
210	8.30	1.314	648.6	7.54	1.302	648.2	6.90	1.291	647.7	6.36	1.282	647.3
220	8.43	1.321	653.9	7.66	1.310	653.5	7.01	1.299	653.0	6.46	1.289	652.6
240	8.69	1.336	664.4	7.89	1.325	664.0	7.22	1.315	663.6	6.66	1.305	663.2
260	8.95	1.351	674.8	8.13	1.340	674.4	7.44	1.329	674.1	6.86	1.320	673.7
280	9.21	1.365	685.2	8.37	1.354	684.9	7.66	1.344	684.6	7.06	1.334	684.3
300	9.47	1.380	695.7	8.60	1.368	695.4	7.87	1.358	695.1	7.26	1.348	694.9

	70 [37.9]			75 [41.3]			80 [44.5]			85 [47.6]		
Sat.	4.12	1.101	548.1	3.86	1.095	548.8	3.63	1.090	549.5	3.43	1.085	550.2
50	4.27	1.117	556.0	3.96	1.107	554.7	3.69	1.097	553.3	3.45	1.088	551.9
60	4.38	1.129	562.4	4.07	1.119	561.1	3.80	1.110	559.9	3.55	1.101	558.6
70	4.49	1.141	568.6	4.18	1.131	567.4	3.90	1.122	566.3	3.65	1.113	565.1
80	4.60	1.152	574.7	4.28	1.143	573.6	4.00	1.134	572.6	3.74	1.125	571.5
90	4.70	1.163	580.7	4.38	1.154	579.7	4.09	1.145	578.7	3.83	1.137	577.7
100	4.81	1.174	586.5	4.48	1.165	585.6	4.18	1.156	584.7	3.92	1.148	583.8
110	4.91	1.184	592.2	4.57	1.175	591.4	4.27	1.166	590.5	4.00	1.158	589.7
120	5.01	1.194	597.9	4.67	1.185	597.1	4.36	1.176	596.3	4.09	1.168	595.6
130	5.11	1.204	603.5	4.76	1.195	602.8	4.45	1.186	602.1	4.17	1.178	601.4
140	5.21	1.213	609.1	4.85	1.204	608.4	4.53	1.196	607.8	4.25	1.188	607.1
150	5.31	1.222	614.6	4.94	1.213	614.0	4.62	1.205	613.4	4.34	1.197	612.7
160	5.41	1.231	620.0	5.04	1.222	619.4	4.71	1.214	618.9	4.42	1.206	618.3
170	5.51	1.240	625.4	5.13	1.231	624.8	4.79	1.223	624.3	4.50	1.215	623.8
180	5.60	1.248	630.8	5.22	1.240	630.3	4.88	1.231	629.8	4.58	1.224	629.3
190	5.70	1.256	636.2	5.31	1.248	635.7	4.96	1.240	635.3	4.66	1.232	634.8
200	5.80	1.265	641.5	5.40	1.256	641.0	5.04	1.248	640.6	4.74	1.240	640.2
210	5.89	1.273	646.9	5.49	1.264	646.4	5.13	1.256	646.0	4.82	1.248	645.6
220	5.98	1.280	652.2	5.58	1.272	651.7	5.22	1.264	651.3	4.90	1.256	650.9
230	6.08	1.288	657.5	5.66	1.280	657.1	5.30	1.272	656.7	4.98	1.264	656.3
240	6.17	1.296	662.8	5.75	1.288	662.4	5.38	1.280	662.0	5.06	1.272	661.6
260	6.36	1.311	673.4	5.93	1.303	673.0	5.55	1.295	672.7	5.22	1.287	672.4
280	6.55	1.325	684.0	6.10	1.317	683.7	5.71	1.309	683.4	5.37	1.302	683.1
300	6.73	1.339	694.6	6.27	1.331	694.4	5.87	1.323	694.1	5.52	1.316	693.8
320	6.91	1.353	705.3	6.44	1.345	705.1	6.04	1.337	704.8	5.68	1.330	704.5

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TABLE 25.—SUPERHEATED AMMONIA (*Continued*)

Pressure	90 [50.5]			95 [53.3]			100 [56.0]			105 [58.6]		
Temp °F.	v	N	H_S	v	N	H_S	v	N	H_S	v	N	H_S
Sat.	3.25	1.080	550.9	3.08	1.075	551.5	2.94	1.071	552.1	2.80	1.067	552.6
60	3.33	1.092	557.4	3.14	1.084	556.1	2.97	1.076	554.9	2.81	1.069	553.6
70	3.42	1.105	564.0	3.23	1.097	562.8	3.05	1.089	561.7	2.89	1.082	560.6
80	3.51	1.117	570.5	3.31	1.109	569.4	3.13	1.102	568.4	2.97	1.095	567.3
90	3.60	1.129	576.8	3.40	1.121	575.8	3.21	1.114	574.8	3.05	1.107	573.8
100	3.69	1.140	582.9	3.48	1.132	582.0	3.29	1.125	581.1	3.13	1.118	580.2
110	3.77	1.150	588.9	3.56	1.143	588.1	3.37	1.136	587.2	3.20	1.129	586.4
120	3.85	1.161	594.9	3.63	1.153	594.1	3.44	1.146	593.3	3.27	1.140	592.5
130	3.93	1.171	600.7	3.71	1.163	600.0	3.52	1.156	599.3	3.34	1.150	598.5
140	4.01	1.180	606.4	3.79	1.173	605.8	3.59	1.166	605.1	3.41	1.160	604.4
150	4.09	1.190	612.1	3.86	1.183	611.5	3.66	1.176	610.8	3.48	1.170	610.2
160	4.17	1.199	617.7	3.94	1.192	617.1	3.73	1.185	616.5	3.54	1.179	616.0
170	4.24	1.208	623.3	4.01	1.201	622.7	3.80	1.194	622.2	3.61	1.188	621.7
180	4.32	1.217	628.8	4.08	1.210	628.3	3.87	1.203	627.8	3.68	1.197	627.3
190	4.40	1.225	634.3	4.16	1.218	633.8	3.94	1.212	633.4	3.74	1.206	632.9
200	4.47	1.233	639.8	4.23	1.226	639.3	4.01	1.220	638.9	3.81	1.214	638.5
210	4.55	1.241	645.2	4.30	1.235	644.8	4.08	1.228	644.4	3.87	1.222	644.0
220	4.62	1.249	650.6	4.37	1.243	650.2	4.15	1.236	649.8	3.94	1.230	649.4
230	4.70	1.257	656.0	4.44	1.251	655.6	4.21	1.244	655.3	4.00	1.238	654.9
240	4.77	1.265	661.3	4.51	1.259	661.0	4.28	1.252	660.7	4.07	1.246	660.4
260	4.92	1.280	672.1	4.65	1.274	671.8	4.41	1.268	671.5	4.19	1.261	671.2
280	5.06	1.295	682.8	4.79	1.288	682.5	4.54	1.282	682.2	4.31	1.276	682.0
300	5.21	1.309	693.5	4.93	1.303	693.2	4.67	1.297	693.0	4.44	1.291	692.8
320	5.36	1.323	704.2	5.07	1.317	704.0	4.80	1.311	703.8	4.56	1.305	703.6
340	5.50	1.337	714.9	5.20	1.330	714.8	4.93	1.324	714.6	4.68	1.318	714.5

	110 [61.1]			115 [63.6]			120 [66.8]			125 [68.1]		
Sat.	2.68	1.063	553.1	2.57	1.059	553.6	2.47	1.056	554.1	2.37	1.052	554.6
70	2.75	1.075	559.4	2.63	1.068	558.3	2.50	1.061	557.1	2.38	1.055	555.9
80	2.83	1.088	566.2	2.70	1.081	565.2	2.57	1.074	564.1	2.45	1.068	563.0
90	2.90	1.100	572.9	2.77	1.093	571.9	2.64	1.087	570.9	2.52	1.081	569.9
100	2.97	1.111	579.3	2.84	1.105	578.4	2.71	1.099	577.5	2.59	1.093	576.6
110	3.04	1.122	585.5	2.90	1.116	584.7	2.77	1.110	583.9	2.65	1.104	583.1
120	3.11	1.133	591.7	2.97	1.127	590.9	2.83	1.121	590.2	2.71	1.115	589.4
130	3.17	1.144	597.8	3.03	1.138	597.0	2.90	1.132	596.3	2.77	1.126	595.6
140	3.24	1.154	603.7	3.10	1.148	603.0	2.96	1.142	602.4	2.83	1.136	601.7
150	3.31	1.164	609.6	3.16	1.158	609.0	3.02	1.152	608.4	2.89	1.146	607.8
160	3.37	1.173	615.4	3.22	1.167	614.8	3.08	1.161	614.3	2.95	1.156	613.7
170	3.43	1.182	621.1	3.28	1.176	620.6	3.14	1.171	620.1	3.01	1.165	619.6
180	3.50	1.191	626.8	3.34	1.185	626.3	3.19	1.180	625.8	3.06	1.174	625.4
190	3.56	1.200	632.4	3.40	1.194	632.0	3.25	1.188	631.5	3.11	1.183	631.1
200	3.62	1.208	638.0	3.46	1.203	637.6	3.31	1.197	637.1	3.17	1.192	636.7
210	3.69	1.216	643.5	3.52	1.211	643.1	3.36	1.205	642.7	3.22	1.200	642.3
220	3.75	1.225	649.0	3.58	1.219	648.6	3.42	1.214	648.2	3.28	1.209	647.9
230	3.81	1.233	654.5	3.64	1.227	654.2	3.48	1.222	653.8	3.34	1.217	653.6
240	3.87	1.241	660.0	3.70	1.235	659.7	3.54	1.230	659.4	3.39	1.225	659.1
260	3.99	1.256	671.0	3.82	1.250	670.7	3.65	1.245	670.4	3.50	1.240	670.2
280	4.11	1.271	681.8	3.93	1.265	681.5	3.76	1.260	681.3	3.60	1.255	681.1
300	4.23	1.285	692.6	4.04	1.280	692.3	3.87	1.275	692.1	3.71	1.270	691.9
320	4.35	1.299	703.4	4.16	1.294	703.2	3.98	1.289	703.0	3.82	1.284	702.8
340	4.47	1.313	714.3	4.27	1.307	714.1	4.09	1.302	713.9	3.92	1.297	713.7
360	4.59	1.326	725.1	4.38	1.321	724.9	4.20	1.316	724.7	4.03	1.311	724.5

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TABLE 25.—SUPERHEATED AMMONIA (*Continued*)

Pressure	130 [70.4]			135 [72.5]			140 [74.5]			145 [76.5]		
Temp. °F.	v	N	H_S	v	N	H_S	v	N	H_S	v	N	H_S
Sat.	2.28	1.049	555.0	2.20	1.046	555.5	2.12	1.043	555.9	2.06	1.040	556.3
80	2.35	1.062	562.0	2.25	1.056	560.9	2.16	1.050	559.9	2.08	1.045	558.9
90	2.42	1.075	568.9	2.32	1.069	567.9	2.23	1.064	566.9	2.14	1.058	566.0
100	2.48	1.087	575.7	2.38	1.081	574.8	2.29	1.076	573.9	2.20	1.071	573.0
110	2.54	1.099	582.3	2.44	1.093	581.5	2.35	1.088	580.7	2.25	1.083	579.9
120	2.60	1.110	588.7	2.50	1.105	587.9	2.40	1.099	587.2	2.30	1.094	586.5
130	2.66	1.121	594.9	2.55	1.115	594.2	2.46	1.110	593.5	2.36	1.105	592.9
140	2.72	1.131	601.1	2.61	1.126	600.4	2.51	1.121	599.8	2.41	1.116	599.2
150	2.77	1.141	607.2	2.66	1.136	606.5	2.56	1.131	605.9	2.46	1.126	605.3
160	2.83	1.151	613.1	2.72	1.146	612.5	2.61	1.141	611.9	2.51	1.136	611.4
170	2.89	1.160	619.0	2.78	1.155	618.4	2.67	1.150	617.9	2.57	1.146	617.4
180	2.94	1.169	624.8	2.83	1.164	624.3	2.72	1.159	623.8	2.62	1.155	623.3
190	2.99	1.178	630.6	2.88	1.173	630.1	2.77	1.168	629.6	2.67	1.164	629.2
200	3.05	1.187	636.2	2.93	1.182	635.8	2.82	1.177	635.3	2.72	1.173	634.9
210	3.10	1.195	641.9	2.98	1.190	641.5	2.87	1.186	641.0	2.76	1.181	640.6
220	3.15	1.204	647.5	3.03	1.199	647.1	2.92	1.194	646.7	2.81	1.190	646.3
230	3.21	1.212	653.2	3.08	1.207	652.8	2.97	1.202	652.4	2.86	1.198	652.0
240	3.26	1.220	658.8	3.13	1.215	658.4	3.01	1.210	658.1	2.90	1.206	657.7
250	3.31	1.227	664.4	3.18	1.223	664.0	3.06	1.218	663.7	2.94	1.214	663.3
260	3.36	1.235	669.9	3.23	1.231	669.6	3.11	1.226	669.2	2.99	1.222	668.9
280	3.47	1.250	680.8	3.33	1.246	680.5	3.20	1.241	680.2	3.08	1.237	679.9
300	3.57	1.265	691.7	3.43	1.260	691.4	3.30	1.256	691.1	3.18	1.251	690.8
320	3.67	1.279	702.6	3.53	1.274	702.3	3.39	1.270	702.0	3.27	1.266	701.7
340	3.77	1.293	713.5	3.62	1.288	713.2	3.48	1.284	713.0	3.36	1.279	712.7
360	3.87	1.306	724.3	3.72	1.301	724.1	3.58	1.297	723.9	3.45	1.293	723.6

	150 [78.5]			155 [80.4]			160 [82.3]			165 [84.1]		
Sat.	1.99	1.038	556.7	1.93	1.035	557.0	1.87	1.032	557.4	1.81	1.030	557.7
90	2.06	1.053	565.0	1.99	1.048	564.0	1.92	1.043	563.1	1.85	1.038	562.1
100	2.12	1.066	572.1	2.04	1.061	571.2	1.97	1.056	570.3	1.90	1.051	569.4
110	2.17	1.078	579.0	2.09	1.073	578.2	2.02	1.068	577.4	1.95	1.063	576.6
120	2.22	1.089	585.7	2.14	1.085	585.0	2.07	1.080	584.2	2.00	1.075	583.4
130	2.27	1.100	592.1	2.19	1.096	591.4	2.12	1.091	590.7	2.05	1.086	590.0
140	2.33	1.111	598.5	2.25	1.107	597.8	2.17	1.102	597.2	2.10	1.097	596.5
150	2.38	1.121	604.7	2.30	1.117	604.1	2.22	1.112	603.5	2.15	1.108	602.9
160	2.43	1.131	610.8	2.35	1.127	610.2	2.27	1.122	609.7	2.20	1.118	609.1
170	2.48	1.141	616.8	2.39	1.137	616.3	2.31	1.132	615.8	2.24	1.128	615.3
180	2.53	1.150	622.8	2.44	1.146	622.3	2.36	1.142	621.8	2.28	1.138	621.3
190	2.57	1.159	628.7	2.48	1.155	628.2	2.40	1.151	627.8	2.32	1.147	627.3
200	2.62	1.168	634.5	2.53	1.164	634.0	2.44	1.160	633.6	2.36	1.156	633.2
210	2.67	1.177	640.2	2.58	1.173	639.8	2.49	1.169	639.4	2.41	1.165	639.0
220	2.71	1.185	645.9	2.62	1.181	645.5	2.53	1.177	645.1	2.45	1.173	644.7
230	2.76	1.194	651.6	2.66	1.189	651.2	2.57	1.185	650.8	2.49	1.181	650.4
240	2.81	1.202	657.3	2.71	1.198	656.9	2.62	1.194	656.5	2.53	1.190	656.1
250	2.85	1.210	662.9	2.75	1.206	662.5	2.66	1.202	662.1	2.58	1.198	661.7
260	2.90	1.217	668.5	2.80	1.213	668.1	2.71	1.209	667.7	2.62	1.205	667.3
280	2.98	1.233	679.6	2.88	1.229	679.2	2.79	1.225	678.9	2.70	1.221	678.5
300	3.07	1.247	690.5	2.97	1.243	690.2	2.87	1.239	689.9	2.78	1.235	689.6
320	3.16	1.261	701.4	3.06	1.257	701.2	2.96	1.254	700.9	2.86	1.250	700.6
340	3.24	1.275	712.4	3.14	1.271	712.2	3.04	1.268	711.9	2.94	1.264	711.7
360	3.33	1.289	723.4	3.22	1.285	723.2	3.12	1.281	722.9	3.02	1.277	722.7
380	3.42	1.302	734.4	3.31	1.298	734.2	3.20	1.294	738.9	3.09	1.290	733.7

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TABLE 25.—SUPERHEATED AMMONIA (*Continued*)

Pressure	170 [85.9]			180 [89.4]			190 [92.7]			200 [95.9]		
Temp °F.	v	N	H_S	v	N	H_S	v	N	H_S	v	N	H_S
Sat.	1.76	1.027	558.1	1.67	1.023	558.8	1.58	1.018	559 4	1.50	1.014	560.0
90	1.78	1.033	561.1	1.67	1.024	559.2	...	.....	....	....	.....	.....
100	1.83	1.046	568.5	1.72	1.037	566 7	1 62	1 028	564 9	1.52	1.020	563.1
110	1.88	1.059	575.7	1.77	1.050	574.0	1 67	1 041	572 3	1.57	1.033	570.7
120	1.93	1.071	582.6	1.81	1 062	581 1	1 71	1.054	579 6	1 61	1.046	578 1
130	1.98	1.082	589.3	1.86	1 074	587 9	1 75	1 066	586 5	1 65	1.058	585 1
140	2.03	1.093	595.8	1.90	1.085	594.5	1 79	1.077	593.1	1 69	1.069	591.7
150	2.08	1 104	602.2	1.94	1 096	600 9	1.83	1.088	599.7	1 73	1 080	598 4
160	2.13	1.114	608.5	1 99	1 106	607 4	1 87	1 098	606 2	1 77	1 091	605 0
170	2.17	1.124	614.7	2.03	1 116	613 6	1 91	1.108	612 5	1 81	1 101	611 4
180	2 21	1.134	620.8	2.07	1 126	619 7	1 95	1 118	618 7	1 85	1 111	617 7
190	2.25	1.143	626.8	2.11	1 135	625 8	1 99	1 128	624.8	1 89	1 121	623 9
200	2.29	1.152	632.7	2.15	1 144	631 8	2 03	1 137	630 9	1.92	1.130	630.0
210	2.33	1.161	638.5	2.19	1 153	637 7	2 07	1 146	636 9	1 96	1 139	636.0
220	2.37	1.169	644.3	2.23	1 162	643 6	2 11	1 155	642 9	1 99	1 148	642 1
230	2.41	1.177	650.0	2.27	1 170	649.3	2 14	1 163	648 6	2 02	1 156	647 9
240	2.45	1.186	655.7	2.31	1 178	655 0	2 18	1 171	654 3	2 06	1 165	653.6
250	2.50	1.194	661.3	2 35	1 186	660.6	2 22	1 179	659.9	2 10	1 173	659.3
260	2.54	1.202	667 0	2.39	1 194	666 3	2 25	1 187	665.7	2 13	1 181	665 1
280	2.61	1.217	678 2	2 46	1 210	677 6	2 32	1 203	677 0	2 20	1 197	676 4
300	2.69	1.232	689.3	2 54	1 225	688 8	2 40	1 218	688 2	2.27	1 212	687 7
320	2.77	1.246	700.3	2.61	1.239	699.8	2 47	1 232	699.3	2.33	1 226	698 8
340	2.85	1 260	711.4	2.69	1.253	710 9	2 54	1 246	710 4	2 40	1 240	709 9
360	2.92	1.274	722.4	2 76	1.267	721 9	2 61	1 260	721 5	2 47	1 254	721 0
380	2.99	1.287	733.4	2 83	1 280	732 9	2 68	1 273	732.5	2 54	1.267	732 1

	210 [98.9]			220 [101 8]			230 [104 7]			240 [107 4]		
Sat.	1.43	1.010	560 5	1.37	1 007	561 0	1 31	1 00	561 5	1 26	1.000	562.0
110	1.48	1.025	569.0	1.40	1 018	567 3	1 33	1 010	565 7	1 27	1 003	564 0
120	1.52	1.038	576 5	1 44	1 031	574 9	1 37	1 024	573 4	1 31	1 017	571 8
130	1.56	1.050	583.6	1.48	1.043	582 1	1 41	1 036	580 7	1 34	1 030	579.3
140	1.60	1.062	590.4	1.52	1.055	589.1	1 45	1.048	587 6	1 38	1 042	586 5
150	1.64	1.073	597.2	1.55	1 066	596.0	1 48	1 060	594 7	1 42	1 053	593 5
160	1.68	1.084	603.8	1 59	1 077	602 7	1 51	1 071	601 6	1 45	1.065	600.4
170	1.72	1.094	610.3	1 63	1.088	609 2	1 55	1 081	608 2	1 48	1 075	607 1
180	1.76	1.104	616.7	1.66	1 098	615 7	1 58	1 092	614 7	1 51	1 086	613 7
190	1.79	1.114	623.0	1.69	1.108	622 0	1 61	1 102	621 0	1 54	1 096	620.1
200	1.82	1.123	629.1	1 72	1.117	628.2	1 64	1 111	627 3	1 57	1 105	626 4
210	1.86	1.133	635.2	1.76	1 126	634 4	1 68	1 120	633.5	1 61	1 115	632 7
220	1.89	1.142	641.3	1.79	1 135	640 5	1 71	1 129	639 6	1 64	1 124	638 8
230	1.92	1.150	647.2	1 82	1.144	646 4	1 74	1 138	645.6	1 67	1 132	644 8
240	1.96	1.159	652.9	1.86	1 152	652 2	1.78	1 146	651 5	1.70	1 141	650.7
250	1.99	1.167	658.6	1.89	1.161	658.0	1 81	1 155	657.3	1 73	1 149	656 6
260	2.02	1.175	664.4	1 92	1.169	663.8	1.84	1 163	663.2	1 76	1 158	662.5
270	2.05	1.183	670.2	1.95	1.177	669.6	1.87	1 171	669 0	1 79	1 166	668 3
280	2.09	1.191	675.8	1 99	1 185	675.2	1.90	1 179	674 6	1.82	1 174	674 0
300	2.15	1.206	687.2	2.05	1 200	686.7	1 96	1 194	686 2	1 87	1.189	685 6
320	2.21	1.220	698.4	2.11	1 214	697 9	2 02	1.209	697 4	1 93	1.203	696 9
340	2.27	1.234	709.5	2.17	1.228	709.1	2 08	1 223	708 6	1 98	1.218	708.1
360	2.34	1.248	720.6	2.23	1.242	720.2	2 14	1 237	719.7	2 04	1 232	719.3
380	2.41	1.261	731.7	2.29	1.256	731.3	2.19	1.250	730.9	2 09	1.245	730.5
400	2.47	1.274	742.8	2.35	1.269	742 4	2.25	1 263	742.0	2.15	1.258	741.7

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TABLE 26.—THERMAL PROPERTIES OF LIQUID AMMONIA

Temp., ° F.	Saturation pressure, lb. per sq. in.	Volume of 1 lb., cu. ft.	Weight of 1 cu. ft., lb.	144 Apv'	Temp., ° F.	Saturation pressure, lb. per sq. in.	Volume of 1 lb., cu. ft.	Weight of 1 cu. ft., lb.	144 Apv'
−110	0.758	0.02202	45.42	0.003	90	181.8	0.02714	36.84	0.92
−105	0.947	.02211	45.23	.004	95	197.3	.02734	36.58	1.00
−100	1.176	0.02220	45.04	0.005	100	213.8	0.02754	36.32	1.09
− 95	1.450	.02229	44.85	.006	105	231.2	.02774	36.06	1.19
− 90	1.778	.02239	44.66	.007	110	249.6	.02795	35.79	1.29
− 85	2.167	.02248	44.47	.009	115	269.2	.02816	35.51	1.40
− 80	2.626	.02258	44.28	.011	120	289.9	.02839	35.23	1.52
− 75	3.164	0.02268	44.09	0.013	125	311.6	0.02862	34.95	1.65
− 70	3.791	.02278	43.89	.016	130	334.6	.02886	34.66	1.79
− 65	4.518	.02288	43.70	.019	135	358.8	.02910	34.36	1.93
− 60	5.358	.02299	43.51	.023	140	384.4	.02936	34.06	2.09
− 55	6.324	.02309	43.31	.027	145	411.3	.02963	33.76	2.26
− 50	7.43	0.02320	43.11	0.032	150	439.5	0.0299	33.45	2.43
− 45	8.69	.02331	42.91	.038	155	469.1	.0302	33.13	2.62
− 40	10.12	.02342	42.71	.044	160	500.1	.0305	32.80	2.82
− 35	11.74	.02353	42.50	.051	165	532.6	.0308	32.47	3.04
− 30	13.56	.02364	42.30	.059	170	566.6	.0312	32.13	3.27
− 25	15.61	0.02376	42.09	0.069	175	602.2	0.0315	31.8	3.51
− 20	17.91	.02388	41.88	.079	180	639.5	.0318	31.5	3.77
− 15	20.46	.02400	41.67	.091	185	678.4	.0322	31.1	4.05
− 10	23.30	.02412	41.46	.104	190	719.0	.0326	30.7	4.34
− 5	26.46	.02424	41.25	.119	195	761.4	.0330	30.3	4.65
0	29.95	0.02437	41.04	0.135	200	805.6	0.0335	29.9	4.99
5	33.79	.02450	40.83	.153	205	851.7	.0340	29.4	5.36
10	38.02	.02463	40.61	.173	210	899.7	.0345	29.0	5.75
15	42.67	.02476	40.39	.196	215	949.6	.0350	28.6	6.16
20	47.75	.02490	40.17	.220	220	1001.4	.0355	28.2	6.59
25	53.30	0.02504	39.95	0.247	225	1055.3	0.0361	27.7	7.1
30	59.35	.02518	39.72	.277	230	1111.3	.0368	27.2	7.6
35	65.91	.02532	39.50	.309	235	1169.5	.0376	26.6	8.1
40	73.03	.02547	39.27	.344	240	1229.9	.0384	26.0	8.7
45	80.75	.02562	39.04	.383	245	1292.5	.0393	25.4	9.4
50	89.1	0.02577	38.81	0.425	250	1357.4	0.0404	24.8	10.2
55	98.0	.02593	38.57	.471	255	1424.7	.0417	24.0	11.0
60	107.7	.02609	38.33	.520	260	1494.4	.0435	23.0	12.0
65	118.1	.02626	38.09	.574	265	1566.6	.0457	21.8	13.3
70	129.2	.02643	37.85	.632	270	1641.3	.0500	20.0	15.2
75	141.1	0.02660	37.60	0.70	273.2	1690.0	0.0678	14.75	21.2
80	153.9	.02678	37.35	.76	....	....	....	.....	.....
85	167.4	.02696	37.10	.84					

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TABLE 27.—PROPERTIES OF SATURATED STEAM (MARKS AND DAVIS)

Absolute Pressure, Pounds per Square Inch.	Temperature, Degrees F.	Heat of the Liquid.	Heat of Vaporisation.	Total Heat.	Heat Equivalent of Internal Work.	Heat Equivalent of External Work.	Entropy of the Liquid.	Entropy of the Vapor.	Total Entropy.	Specific Volume.	Density Weight per Cubic Foot, Pounds.
p	t	h	L	$H = h + L$	ρ	$Ap\mu$	θ	$\frac{L}{T}$	$\theta + \frac{L}{T}$	v	w
†0.1	35 03	3.05	1071.7	1074.7	1017.3	54.4	0.0062	2.1666	2.1728	2935.0	0.000340
†0.2	53.15	21.23	1061.6	1082.8	1005.2	56.5	0.0423	2.0704	2.1127	1524.0	0.000656
†0.3	64.49	32.57	1055.3	1087.9	997.7	57.6	0.0640	2.0135	2.0775	1041.0	0.000961
0.4	72.91	40.95	1050.6	1091.6	992.4	58.5	0.0800	1.9730	2.0530	794.0	0.001259
0.5	79.68	47.71	1047.0	1094.6	987.6	59.3	0.0926	1.9413	2.0332	642.0	0.001555
0.6	85.32	53.34	1043.8	1097.1	983.9	59.9	0.1029	1.9155	2.0184	541.0	0.001850
0.7	90.18	58.18	1041.1	1099.3	980.7	60.4	0.1117	1.8936	2.0053	467 0	0.002143
0 8	94.46	62.45	1038.7	1101.2	977.8	61.0	0.1195	1.8747	1.9942	412.0	0.002431
0.9	98.33	66.31	1036.6	1102.9	975.2	61.4	0.1265	1.8578	1.9843	367.9	0.002719
1	101.83	69.8	1034.6	1104.4	972.9	61.7	0.1327	1.8427	1.9754	333.0	0.00300
2	126.15	94.0	1021.0	1115.0	956.7	64.3	0.1749	1.7431	1.9180	173.5	0.00576
3	141.52	109.4	1012.3	1121.6	946.4	65.8	0.2008	1.6840	1.8848	118.5	0.00845
4	153.01	120.9	1005.7	1126.5	938.6	67.0	0.2198	1.6416	1.8614	90.5	0.01107
5	162.28	130 1	1000.3	1130.5	932.4	68.0	0.2348	1.6084	1.8432	73.33	0.01364
6	170.06	137.9	995.8	1133.7	927.0	68.8	0.2571	1.5814	1.8285	71.89	0.01616
7	176.85	144.7	991.8	1136.5	922.4	69.4	0 2579	1.5582	1.8161	53.56	0.01867
8	182.86	150.8	988.2	1139.0	918.2	70.0	0 2673	1.5380	1.8053	47.27	0.02115
9	188.27	156.2	985.0	1141.1	914.4	70.6	0.2756	1.5202	1.7958	42.36	0.02361
10	193.22	161.1	982.0	1143.1	910.9	71.1	0.2832	1.5042	1.7874	38.38	0.02606
11	197.75	165.7	979.2	1144.9	907.8	71.5	0.2902	1.4895	1.7797	35.10	0.02849
12	201.96	169.9	976.6	1146.5	904.8	71.8	0.2967	1.4760	1.7727	32.36	0.03090

* Courtesy of the Publishers, Longmans, Green & Co. † Interpolated.

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TABLE 27.—PROPERTIES OF SATURATED STEAM (*Continued*)

Absolute Pressure, Pounds per Square Inch	Temperature, Degrees F.	Heat of the Liquid.	Heat of Vaporization.	Total Heat.	Heat Equivalent of Internal Work.	Heat Equivalent of External Work.	Entropy of the Liquid.	Entropy of the Vapor.	Total Entropy.	Specific Volume.	Density Weight per Cubic Foot, Pounds.
p	t	h	L	$H = h + L$	ρ	Apu	θ	$\frac{L}{T}$	$\theta + \frac{L}{T}$	v	w
13	205.87	173.8	974.2	1148.0	902.0	72.2	0.3025	1.4639	1.7664	30.03	0.03330
14	209.55	177.5	971.9	1149.4	899.3	72.6	0.3081	1.4523	1.7604	28.02	0.03569
14.7	212.00	180.0	970.4	1150.4	897.6	72.9	0.3118	1.4447	1.7565	26.79	0.03732
15	213.0	181.0	969.7	1150.7	896.8	72.9	0.3133	1.4416	1.7549	26.27	0.03806
20	228.0	196.1	960.0	1156.2	885.8	74.3	0.3355	1.3965	1.7320	20.08	0.04980
25	240.1	208.4	952.0	1160.4	876.8	75.3	0.3532	1.3604	1.7136	16.30	0.0614
30	250.3	218.8	945.1	1163.9	869.0	76.2	0.3680	1.3311	1.6991	13.74	0.0728
35	259.3	227.9	938.9	1166.8	862.1	76.9	0.3868	1.3060	1.6868	11.89	0.0841
40	267.3	236.1	933.3	1169.4	855.9	77.6	0.3920	1.2841	1.6761	10.49	0.0953
45	274.5	243.4	928.2	1171.6	850.3	78.1	0.4021	1.2644	1.6665	9.39	0.1065
50	281.0	250.1	923.5	1173.6	845.0	78.6	0.4113	1.2468	1.6581	8.51	0.1175
55	287.1	256.3	919.0	1175.4	840.2	78.9	0.4196	1.2309	1.6505	7.78	0.1285
60	292.7	262.1	914.9	1177.0	835.6	79.7	0.4272	1.2160	1.6432	7.17	0.1394
65	298.0	265.7	911.0	1178.5	831.4	79.8	0.4344	1.2034	1.6368	6.65	0.1503
70	302.9	272.6	907.2	1179.8	827.3	80.1	0.4411	1.1896	1.6307	6.20	0.1612
75	307.6	277.4	903.7	1181.8	823.5	80.5	0.4474	1.1778	1.6252	5.81	0.1721
80	312.0	282.0	900.3	1182.3	819.8	80.7	0.4535	1.1665	1.6200	5.47	0.1829
85	316.3	286.3	897.1	1183.4	816.3	81.0	0.4590	1.1561	1.6151	5.16	0.1937
90	320.3	290.5	893.9	1184.4	813.0	81.2	0.4644	1.1461	1.6105	4.89	0.2044
95	324.0	294.5	890.9	1185.4	809.7	81.5	0.4694	1.1367	1.6061	4.65	0.2151
100	327.8	298.3	888.0	1186.3	806.6	81.7	0.4743	1.1277	1.6020	4.429	0.2258

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TABLE 27.—PROPERTIES OF SATURATED STEAM (*Continued*)

105	331.4	302.0	885.2	1187.2	803.6	81.9	0.4789	1.1191	1.5980	4.230	0.2635
110	334.8	305.5	882.5	1188.0	800.7	82.1	0.4834	1.1108	1.5942	4.047	0.2472
115	338.1	309.0	879.8	1188.8	797.9	82.3	0.4877	1.1030	1.5907	3.880	0.2577
120	341.3	312.3	877.2	1189.6	795.2	82.5	0.4919	1.0954	1.5873	3.726	0.2683
125	344.4	315.5	874.7	1190.3	792.6	82.6	0.4959	1.0880	1.5839	3.583	0.2791
130	347.4	318.6	872.3	1191.0	790.0	82.8	0.4998	1.0809	1.5807	3.452	0.2897
135	350.3	321.7	869.9	1191.6	787.5	82.9	0.5035	1.0742	1.5777	3.331	0.3002
140	353.1	324.6	867.6	1192.2	785.0	83.0	0.5072	1.0675	1.5747	3.219	0.3107
145	355.8	327.4	865.4	1192.8	782.7	83.2	0.5107	1.0612	1.5719	3.112	0.3213
150	358.5	330.2	863.2	1193.4	780.4	83.3	0.5142	1.0550	1.5692	3.012	0.3320
155	361.0	332.9	861.0	1194.0	778.1	83.5	0.5175	1.0489	1.5664	2.920	0.3425
160	363.6	335.6	858.8	1194.5	775.8	83.6	0.5208	1.0431	1.5639	2.834	0.3529
165	366.0	338.2	856.8	1195.0	773.6	83.7	0.5239	1.0376	1.5615	2.753	0.3633
170	368.5	340.7	854.7	1195.4	771.5	83.8	0.5269	1.0321	1.5590	2.675	0.3738
175	370.8	343.2	852.7	1195.9	769.4	83.9	0.5299	1.0268	1.5567	2.602	0.3843
180	373.1	345.6	850.8	1196.4	767.4	84.0	0.5328	1.0215	1.5543	2.533	0.3948
185	375.4	348.0	848.8	1196.8	765.4	84.1	0.5356	1.0164	1.5520	2.468	0.4052
190	377.6	350.4	846.9	1197.3	763.4	84.2	0.5384	1.0114	1.5498	2.406	0.4157
195	379.8	352.7	845.0	1197.7	761.4	84.3	0.5410	1.0066	1.5476	2.346	0.4262
200	381.9	354.9	843.2	1198.1	759.5	84.4	0.5437	1.0019	1.5456	2.290	0.437
205	384.0	357.1	841.4	1198.5	757.6	84.5	0.5463	0.9973	1.5436	2.237	0.447
210	386.0	359.2	839.6	1198.8	755.8	84.5	0.5488	0.9928	1.5416	2.187	0.457
215	388.0	361.4	837.9	1199.2	754.0	84.6	0.5513	0.9885	1.5398	2.138	0.468
220	389.9	363.4	836.2	1199.6	752.3	84.7	0.5538	0.9841	1.5379	2.091	0.478
225	391.9	365.5	834.4	1199.9	750.5	84.7	0.5562	0.9799	1.5361	2.046	0.489
230	393.8	367.5	832.8	1200.2	748.8	84.8	0.5586	0.9758	1.5341	2.004	0.499
240	397.4	371.4	829.5	1200.9	745.4	85.0	0.5633	0.9676	1.5309	1.924	0.520
250	401.1	375.2	826.3	1201.5	742.0	85.1	0.5676	0.9600	1.5276	1.850	0.541
275	409.5	384.2	818.6	1202.8	734.2	85.3	0.5780	0.9419	1.5199	1.686	0.593
300	417.5	392.7	811.3	1204.1	726.8	85.6	0.5878	0.9251	1.5129	1.551	0.645

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